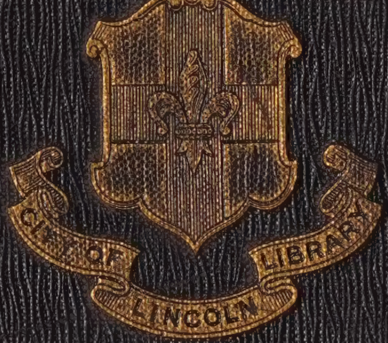
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THE DESIGN AND CONSTRUCTION

OF

INTERNAL-COMBUSTION ENGINES

A HANDBOOK FOR

DESIGNERS AND BUILDERS OF GAS AND OIL ENGINES

BY HUGO GÜLDNER
Chief Engineer and Director, Güldner-Motoren-gesellschaft, Aschaffenburg

TRANSLATED FROM THE SECOND REVISED EDITION

WITH ADDITIONS ON

AMERICAN ENGINES

BY

H. DIEDERICHS
Professor of Experimental Engineering
Cornell University

728 Illustrations and 36 Folding Plates

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AUTHOR'S PREFACE TO THE FIRST EDITION

GERMANY'S gas engine industry justly enjoys an international reputation. Whether judged by the standards of age and experience, commercial importance or originality of design, it leads the world. Four-cycle and two-cycle engine, explosion and constant-pressure engine, automobile construction, and commercial utilization of blast-furnace gas —in short, everything that has served to lay the foundation of the industry and that has helped to make it vital and important is either the product of German thought, or was first practically realized on German soil.

Our technical literature on the subject shows the same high degree of development. There are a number of important books as well as occasional articles treating of special fields of the industry, which for thoroughness and depth of scientific treatment hardly find their equal. But as far as the industry itself is concerned such treatises and publications are of little real and practical use. Writers are apt to say a good deal more about the general construction of *existing* internal-combustion engines than about the details of the problems involved, and as a matter of fact they generally get more information *from* the practical designer than they give *to* him. This objection holds against most of the technical handbooks now existing. Outside of the thermodynamic and thermal principles involved, at best so arranged as to serve a real purpose, such books very often contain nothing more than a series of descriptions of existing engines, illustrated by figures giving little idea of proportion of parts, and a few results of tests. Of how much real use are such works to the practical man in the field? Taking it altogether he usually finds what to him is ancient history, and whatever is new to him is often presented in such form as to make it unavailable to his needs. The trouble is that both text and illustration are arranged to suit the layman, and consequently the builder or designer finds little of practical value. Thus it happens that our present handbooks on internal-combustion engines have but little circulation among the men in the field, but are used mostly by those who first wish to study the nature of the internal-combustion engine and for that purpose require a more or less popularly written general treatise.

The reasons for this condition of affairs are not far to seek. The gas engine originated and grew through experimentation and experience. All the improvements that were made and are still being made are the result of experimentation, and only in the course of the latter do the men actually engaged in the development of the industry acquire their special knowledge and experience. New developments in the industry are presented to outsiders usually in the perfected state, and if after such presentation scientific investigation takes hold of the subject, it—with very few exceptions—concerns itself mainly with the theoretical thermodynamic facts and principles involved. To thus confine attention to a field, which is really only a small part of the engine constructor's actual work, and which is in reality of much less importance than is commonly assigned to it, simply amounts to a failure on the part of scientific men to appreciate a real need.

The author desires not to be misunderstood in making the above statements. No one doubts for a moment that thorough tests made on internal-combustion engines are of the highest scientific importance, besides possessing an indirect practical value, although the final results of such investigations are generally of a purely theoretical, and often even of hypothetical, nature and can consequently be practically utilized only in isolated instances. The criticisms made are not aimed at scientific investigation of the thermal problems involved, but at the *highly exaggerated valuation* put upon it as an auxiliary in gas engine design and construction, and at the *narrow viewpoint* which this tendency has succeeded in impressing upon all experimental work connected with the gas engine. The latter effect is strongly manifest in our technical literature as well as in our technical instruction. For some decades past science has apparently considered only the *physical* and *thermal* phenomena connected with the gas engine as worthy of its attention, and, in doing so, has seriously neglected the investigation of the *static* and *kinetic* features involved, that is, has paid little attention to the *theory of the design* of the internal-combustion engine. Complete volumes are filled with thermodynamic discussions, while a few pages will cover all that is written concerning constructive and technical details.

There is little doubt that these things have constitiuted a serious drawback in the development of the industry and have worked considerable harm. For, even to-day, each manufacturing concern is compelled to determine for itself the best forms, the best mode of action and the allowable safe stresses for each essentially new detail of gas-engine construction, and is not able in any way to avail itself of the results of investigations made and of the experiences gained in probably a large number of similar cases in other works. As a consequence, many futile experiments are repeated again and again. The continued great loss of time and money incident to this proceeding could at least in large part be with certainty avoided if scientific investigation, instead of almost entirely confining itself to the *thermodynamic* side of the problem, could be induced to take up with equal energy the *mechanical* and *kinetic* side with special reference to the *problems* of *actual design* and *practical construction.* The solution of the entire problem does not rest entirely in the domain of thermodynamics; on the contrary, every technically trained man and every practically trained mechanic can do his share toward it. Under such conditions, technical literature, in so far as it pertains to the gas engine, would soon assume a quite different aspect and would attain a higher degree of importance in the practical field than it at present possesses.

It is the aim of this book to make a beginning at clearing the ground along the lines thus defined and to be considered as the first modest contribution to this cause. Out of the fund of practical experience, gathered during the past fifteen years in positions of executive capacity, it has been written by a gas-engine designer and builder for others engaged in the same work, with the idea of presenting to the technical man, and especially to the gas-engine designer, a reliable handbook and to the gas-engine builder new in the field a practically useful guide. That there is real need for a book of this kind for both classes of men, the author believes to have shown in what has been said.

Concerning the scope and manner of treatment of the material entering into the various problems, the book may speak for itself. The entire internal arrangement was in reality already fixed by the purpose for which the work is written.

The *first* part[1] of the book consists of a critical discussion of the most important of the older types of internal-combustion engines, to serves both as an introduction to the field in general and to give a brief résumé concerning constructions already tried and in part abandoned.

The *second* part treats first of the thermodynamics of the gas engine in a form and to an extent which the real purpose of the book seems to justify. This is followed by a critical examination of the various events of the gas-engine cycles, a subject which should be welcomed alike by the designing-room and the shop and which, on account of its importance, is quite likely to receive further elucidation and additions at the hands of others.

The most extensive part of the book, however, the one which brings out most clearly the real aim of the entire work, is the *third* part, intended to serve as an everyday working guide to the designer and constructor. The author would like to look upon this as the beginning of a "science of gas-engine construction" whose aim it shall be to suitably extend the rules of general design and to adapt them to the special requirements of gas-engine construction, to bring together a selected collection of dependable data, and of approved forms of construction to serve as a reliable guide for new constructions, and thus save both designer and builder a great deal of unnecessary and costly trying-out and experimentation. This aim calls for the most extended use of drawings, true to detail, showing approved designs, and the more dimension figures are given in them, the better. The dimensions of machine parts tried and found satisfactory in actual service constitute statistics of peculiar value to the designer, and not infrequently such figures express rules of design not yet stated clearly in words. In fact, in cases where the underlying rules and laws are not yet clearly understood, such combinations of drawing and dimension find their most important usefulness. This viewpoint entirely determined the manner of illustrating the book, and especially of the third part of the same. If this practice should serve as an example followed by others, it can only prove of benefit to technical gas-engine literature, in general, because the hitherto much preferred method of banishing all dimension figures from text illustration has resulted in merely turning these industrial documents into useless pictures.

While the third part shows the method of designing the various separate machine parts, it is the main purpose of the *fourth* part to show how these various parts are utilized in combination in the modern types of internal-combustion engines and to give some information concerning the field of application, erection, etc., of the various forms. The results of the latest tests also reported in this part serve to show the present economic position of the gas engine and its standing as compared with other sources of power.

The *fifth* part, treating of the gas-engine fuels and combustion in the gas engine, is also mainly arranged to suit the needs of the designer, but there is little doubt that the shopman will also find many things of service to him. Even engine builders grown gray in the business are sometimes not entirely clear on the various phenomena attending combustion in gas engines, nor how important all means tending to promote complete combustion really are and what means are available. It requires a study of the combustion process to see that, in a certain sense, air is power as far as the gas engine is concerned, and that to neglect this fact simply means a waste of fuel.

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[1] TRANSLATOR'S NOTE. The parts of the book have been rearranged, see Translator's Preface.

In order that the designer and builder not familiar with thermodynamics may find the book entirely useful, a brief treatise of Thermodynamics and Thermochemistry is given in the Appendix, to avoid complicating the main part of the book with elementary discussions. The rest of the subject matter of the Appendix is a miscellaneous collection of information on several topics, which, while it will probably not be regularly used, will now and then prove of interest and value.

Taking it altogether, this work is written "out of practice for practice;" it is nevetheless hoped that it will be of some use in technical instruction. For obvious reasons, the author, in making this statement, has in mind mainly the technical high school. The training which these schools at present impart to the engineering graduate along the line of gas-engine design amounts to next to nothing. Every chief designer can corroborate this statement from his own experience, no matter from where his assistants may come. That this condition of affairs should change is merely an industrial necessity and consequently a just demand on the part of the gas-engine industry.

The objection to this statement of the case is the old one that "the high school should not be called upon to develop specialists." How long this view of the question can be maintained in the face of the steady readjustment now at work in the industrial field is problematical. What is not open to question is the fact that the gas-engine industry has to-day attained an extent and importance which make it barely secondary to steam-engine and machine-tool industry and to the electrical manufacturing interests. Now it is well known that the technical high schools in their present courses of study, examinations, and laboratory equipments, try to meet the practical requirements of these branches of our industry to the widest possible extent. It is therefore strongly in the interest of the gas-engine builder to see that those students who intend later to specialize in his branch of industrial activity, are given sufficient opportunity to acquire at least the fundamentals necessary to this end. There seems to be an impression that the special knowledge required can be imparted simply by a study of comparative machine design and of thermodynamics. The fact is that the origin of most of the special training required goes back to theoretical machine design and to the practical design problems and exercises connected therewith. Both courses of study therefore urgently call for adaptation to the proven need. With this aim in view it is hoped that the present work may in a measure serve as a guide and that its many examples and problems may furnish welcome material for practice in the drafting-room.

AUGSBURG, December, 1902.

AUTHOR'S PREFACE TO THE SECOND EDITION

The good reception accorded the first edition of this handbook compelled the preparation of a second edition as far back as June, 1903, only a few months after the first appearance of the book. On account of business engagements, however, the revision for the edition was delayed a very considerable time, although the delay had one good feature in that it allowed of the insertion of the latest developments in the gas-engine field. The last few years especially have brought forth not a little of what is new and of general importance in the construction of gas engines.

The *gas turbine*, more than one hundred years after its first appearance, has again stepped into the foreground and is again the subject of energetic activity of investigators and inventors, who are apparently blinded by the rapid commercial success of the steam turbine.

Suction gas producers, constituting the simplest and cheapest of power installations, are in a fair way of completely displacing the older types of gas producers and are even making considerable inroads into the field hitherto reserved to the smaller illuminating gas and oil engines. This development has incidentally been the cause of the taking up of the most important problem yet remaining in the gas-engine field, that of the gasification of tar-carrying fuels. This problem is now receiving great attention and many fairly successful solutions have already appeared.

The development and introduction of the *double-acting principle* in the building of 4-cycle engines has given the latter type of machine a new lease of life in the struggle between it and the 2-cycle engine for supremacy in the field of large engines. The conditions are further improved in favor of the 4-cycle engine by substituting for the old *cam-operated gear* an *eccentric valve gear*, in part so designed as to operate the valves positively throughout. More perfect methods of *speed regulation* for this type of engine have also been devised.

In this manner one improvement has followed another since the first appearance of this bock and this active development of the internal-combustion engine was duly taken into account in the preparation of the second edition. As a consequence, Part III, treating of the theoretical design of the various engine parts, and Part IV, dealing with modern types of engines, have been considerably enlarged. In order to keep the book from becoming unwieldy on account of the necessary insertions, the historical discussion of Part I[1] has in this edition been cut down to include only the distinct and original types of internal-combustion engines and this part has thus been shortened in spite of the fact that a brief discussion on gas turbines has been inserted. As far as the rest of the main divisions are concerned, no reduction of any importance proved possible, in fact, in most of them many, and in some cases extended, insertions were urgently called for.

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[1] This part is entirely omitted from the translation.

For the details of the revision the reader is referred to the Table of Contents and to the Index; it will be sufficient merely to mention that the number of pages has been increased by 80, the number of illustrations in the text by 50, the number of plates by 18, and the number of tables by 13.

Of course neither the inherent character nor the clearly defined aim of the book have been changed in any manner. As in the past, its one guiding principle is "*less invention, more rational design*," in other words, let us have less guesswork, less rule-of-thumb and more of sane design based on tried principles. And of hardly less importance is the second plea to *establish first a sound working basis for rational design before proceeding to a lot of hypothetical thermodynamic experimentation.*

To the first of these demands there are probably few dissenting voices, except those of professional inventors. There is, however, no such unanimity of opinion regarding the second, which is mainly aimed at our present-day method of technical instruction and at scientific investigation. It came as a surprise to note that some of the strongest objections to this demand came out of the camp of the practical engine builder, while one of the most eminent of our university professors has turned into a most energetic champion of the cause opposed to mere theorizing.

The opinions which the author himself holds concerning one or the other of the requirements or demands above laid down, have been clearly set forth in the preface to the first edition. The latter expresses a warning against the over-valuation of one-sided theoretical investigations and merely suggested that a study of the mechanical and kinetic laws underlying the construction of the gas engine be in future preferred to the former type of investigation. That, however, is by no means equivalent to a condemnation of thermodynamic study in general, much less does it amount to an expression of disrespect aimed at the authorities in that field of knowledge, as several of the reviews of the first edition were pleased to state. The author desires to leave no doubt in any one's mind that this interpretation does not represent his own ideas on the subject. The theory of gas-engine construction is indebted to thermodynamics for a great many things and at all times is dependent upon its coöperation,—but certainly not to the exclusion of everything else.

The last words are due also in this case to the colleagues and firms who have kindly aided in the preparation of this second edition and who have given of their time sometimes at a loss to their own material advantages. To all of them the author hereby expresses his thanks, hoping that he may have their continued coöperation.

MÜNCHEN, June, 1905.

TRANSLATOR'S PREFACE

In adding this work to the technical gas engine literature already existing in this country, the translator feels it hardly necessary to give any extended reasons for his belief that the book will fill a real want. If any justification for its appearance should be thought necessary, the reader is referred to the author's preface to the first edition. The statements there made concerning the state of technical gas engine literature in Germany apply with equal force to American conditions. As a matter of fact, outside of a considerable fund of practical information concerning the automobile engine, there is very little data available in our technical literature treating of the design and construction of stationary engines, large or small. This fact has been brought home to the translator in a number of instances, both in his capacity as teacher and as consulting engineer or designer. It is by no means intended to convey the impression that American gas-engine industry is behind the times or that rules-of-thumb are supreme in its designing rooms. Detailed practical data and information no doubt exist here as well as abroad, but they have usually been acquired by hard work in the way of experimentation at a considerable expenditure of time and money, and are conseqently in most cases considered a valuable asset of the business not to be published broadcast. There is no doubt that this procedure is entirely justified from the standpoint of the manufacturers, but, as Güldner has pointed out, it leads to a large amount of useless investigation, results in the appearance on the market of freak forms which cannot maintain themselves for any length of time, and, finally, it serves to retard standardization of practice.

It remains to point out in what particulars the book as here presented differs from the scope and contents of the original. In the first place the translation is quite free without losing the sense of the original. In the attempt to avoid the stilted expressions sometimes found in translations, the text was read by at least two engineers little conversant with German, and while it is of course quite impossible to avoid them altogether, it is believed that so-called "Germanisms" are at least of infrequent occurrence. In order to make the book more acceptable to those engineers and designers who have had no opportunity to become familiar with the metric system of units, all of the figures in both text and illustrations have been transposed from metric to English units, except perhaps in a few isolated instances where the system of units made no particular difference. To avoid confusing the text, the original metric dimensions and figures have not been repeated, as is sometimes the practice. Again, there are a few exceptions to this statement in cases where a juxtaposition of the figures in the two systems seemed important. The transposition from millimeters and centimeters to inches is not in a round-numbered ratio, and since in many of the computations and illustrations it seemed important to retain the exact dimensions, inches will in most cases be found expressed to the second decimal place, that is, in hundredths. This may at first seem strange to the American engineer or mechanic for whom the smallest division is usually $\frac{1}{64}$ of an inch. But as stated, this method of transposition was necessary in many of the problems and cuts, and for the sake of consistency it was followed throughout the book. It is not believed that the value of either the problem or the drawing is impaired thereby.

Perhaps the most radical difference from the original consists in the omission of the entire first part treating of the history of the gas engine. Thus Part II of the German work has become Part I of the American edition, etc. Although Güldner's treatment of the historical development of the gas engine is quite original and valuable on account of the citation of many early test results, it was felt that the same subject had been so well treated by English writers like Clerk, Donkin, and Robinson, that this particular division of the German edition could be omitted without great detriment to the rest of the book. A second reason which contributed to this decision was the fact that in order to make the book more acceptable to the American reader in general it became necessary to add a somewhat extended discussion of the important machines on the American market. This insert largely replaces the pages lost by the omission of Part I of the original and, without this omission, the book would undoubtedly have become somewhat unwieldy.

The translator is fully aware that the subject matter of the insert on American engines is not in any respect up to the standard set by Güldner in his discussion of German machines. In simple justice to himself he is compelled to state that, with a few exceptions, this is due to the extreme reticence on the part of many manufacturers, as already mentioned, to give out any information concerning their engines, and is in no sense due to any neglect on his part to get the best information available.

In conclusion, the translator wishes to express his sincere thanks to those firms who have met to the fullest extent his requests for information and to those of his colleagues who have given their time without stint to the revision of manuscript and text. Special mention is due to Dr. F. E. Junge of Berlin, who first suggested the desirability of undertaking the work and to whose help is due a large share of the accomplishment of the same; also to Mr. C. F. Hirshfeld, Professor of Gas Engineering in Cornell University and to Mr. A. G. Kessler, Instructor in Gas Engineering, for reading the manuscript and the text, and to Mr. G. W. Lewis, Instructor in Machine Design, for his efficient work in preparing the drawings.

ITHACA, N. Y., August 16, 1909.

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PART I

THE VARIOUS METHODS OF OPERATING GAS ENGINES AND THE GAS ENGINE CYCLES

A. GENERAL CONSIDERATIONS

I. Classification of Engines

1. The term Internal-combustion Engine is applied to all prime movers employing gaseous or liquid fuel, that is, to all gas and oil engines without regard to the cycle of operation. Using the cycle as a basis of classification, we have **Constant-volume Engines** in which the charge burns suddenly and at approximately constant volume (Lenoir and Otto), and **Constant-pressure Engines** in which the charge burns gradually and at approximately constant pressure (Brayton and Diesel).[1]

2. It is not correct to designate a double-acting or multi-cylinder 4-cycle engine by the term "2-cycle," because the expressions "4-cycle" and "2-cycle" state the number of strokes required to complete one cycle on one side of the piston. Combining cylinders or making one cylinder double-acting in the case of a 4-cycle engine in no way changes the number of strokes to each working cycle.[2] As far as the strokes required per cycle are concerned, we therefore distinguish only two classes of engines:

Single-cylinder, single- or double-acting, or multi-cylinder, single- or double-acting, **4-cycle engines**; and

Single-cylinder, single- or double-acting, or multi-cylinder, single- or double-acting, **2-cycle engines.**

---

[1] Translator's Note. In German practice, gas engines are sometimes classified as "explosion" and "combustion" engines, the former referring to constant-volume, the latter to constant-pressure engines. Güldner points out that the classification is not good, for explosion engines are combustion engines. He further makes the point that the designation "explosion" engine given to those employing the Otto cycle does not exactly meet the case, taking the ground that explosive combustion is much more rapid than that occurring in Otto gas engines in ordinary operation. He proposes the term "Verpuffung" instead of explosion, meaning rapid combustion but falling short of the nature of an explosion; but since the English language does not contain an equivalent term, the name explosion engine will have to stand as an alternative for "engines with combustion at constant volume."

Güldner objects to this term as well as to the designation "engines with combustion at constant pressure," because of their length; hence the contraction of these expressions into those used in the above paragraph.

[2] Translator's Note. For the same reason, Güldner objects to the term "Ein-Takt," meaning "one-stroke cycle," which is often applied in Germany to the double-acting 2-cycle engine. This term has never to the writer's knowledge been used in the United States.

3. The following distinctions, regarding cylinder combinations, etc., should also be drawn at this time:

As opposed to **single-cylinder engines,** those having two or more cylinders are known as **multi-cylinder engines.** The latter are subdivided into **tandem engines,** when the cylinders are arranged in a line one behind the other, and **double** or **twin** (two-cylinder, three-cylinder, four-cylinder, etc.) engines when the cylinders are arranged side by side. For other terms designating multi-cylinder engines, see Part II, p. 63.

Distinguished from **horizontal engines,** we have the two forms of **vertical engines,** one with the crank-shaft below, and the other with the shaft above the cylinder. As usually constructed, horizontal engines are **right-handed** and **run-over,** that is, standing on the lay shaft side of the engine, the crank will be found on the right-hand side, and the direction of rotation is clockwise as viewed from this side. In vertical engines, these distinctions are not so clear. Facing what may be termed the front side of the engine (the side on which most of the valve gear and the adjustments are found) the wheel is usually at the right and the top of the wheel moves toward the spectator. This arrangement, however, is by no means strictly adhered to.

II. Thermodynamic Definitions

1. The work equivalent of the heat unit (B.T.U.) is taken at **778** foot-pounds, so that the heat equivalent of work or the **mechanical heat equivalent** is

$$A=\frac{1}{778}=.001285 \text{ British Thermal Units.}$$

Many writers still use Joule's equivalent (772) instead of 778, but the recent investigations of Dietrich, among others, admit of but little doubt that the true value of the heat equivalent is **higher,** near 783.[1] It is consequently more nearly correct to employ the higher of the two values now in common use. With $A=\frac{1}{778}$ B.T.U., thermal efficiency computations gives results .935%, that is, less than 1% smaller than if $A=\frac{1}{772}$ B.T.U. were used.

2. The specific heats c_p and c_v may be considered ***constant*** **for all pressures and temperatures** occurring in internal-combustion engine practice. This assumption is at least safer than the use of the formulæ proposed by Mallard and Le Chatelier, Table 1. According to these, for example, for a temperature, range from 32 to 3632° F., c_v for air increases from .1666 to .2506 (i.e., 1:1.51), while for carbon dioxide the increase is even greater, from .1422 to .3091 (1:2.18).

Now recent unimpeachable investigations have seriously shaken faith in the accuracy of the above formulæ and in the trustworthiness of the experiments from which they were derived. Fliegner[2] especially, from his own recent experiments, tersely concludes "that the results of Mallard and Le Chatelier may even be looked upon as proof that the specific heat of gases at constant volume does not appreciably change at high temperatures, say up to 3600°." Zeuner,[3] holding the same view, says: "If we were to accept the results of Mallard and Le Chatelier as correct, nearly everything at present accepted or considered established in connection with high temperature computation and with the phenomena of the internal-combustion engine would fall. In establishing their empirical formulæ, these engineers used only very few tests at high temperatures (up to 3600° F.) of their *own*, and the trend of their conclusions is therefore at least open to question."

---

[1] Compare, for example, Zeuner, Thermodynamik, 2 ed., Part I, pp. 13 and 120. The International Congress of Electrical Engineers in 1904 fixed upon $A=1/777$ B.T.U.

[2] Vierteljahrschrift der Naturforschenden Gesellschaft in Zürich, 1899, p. 192.

[3] Zeuner, Thermodynamik, 2 ed., Part I, p. 143.

Eugene Meyer,[1] in his "Untersuchungen am Gasmotor," comes to the general conclusion that, while the specific heats are not constant, they do not increase with temperature as rapidly as stated by Mallard and Le Chatelier.

A similar stand is taken by Arnold Langen,[2] who, on the basis of numerous scientific tests on the explosion pressures of hydrogen and carbon monoxide mixtures in closed vessels, derives the following equations:

For simple (di-atomic) gases, $c_v = 4.8 + .0006t$
For water vapor (H_2O), $c_v = 5.9 + .00215t$
For carbon dioxide (CO_2), $c_v = 6.7 + .00260t$

(Note that t in this case is temperature in degrees C.)

The final solution of the question *whether or not specific heats vary with temperature*, is of interest not only to the student of thermodynamics, but also, in a high degree, to the designer of gas engines. Upon this solution, for example, depends our ability to compute the temperatures occurring in the gas engine-cylinder, to determine the energy of the working medium going through the cycle, and finally to give some idea of the relation between the energy so determined and the various energy losses. As the case stands at present, and as long as the question of the variation of specific heat with temperature is not definitely settled, we are not even able to state positively whether the losses due to the imperfections of our present-day engines are relatively large or small, or whether, by improving fuel mixtures and ignition, perfecting combustion in general, we will be able to raise the thermal efficiency much or little.

Table 1

VARIATION OF SPECIFIC HEATS ACCORDING TO MALLARD AND LE CHATELIER

		$c_p =$	$c_v =$
Air		$\begin{cases} .2360 + .000023t \\ .2245 + .000023T \end{cases}$	$.1666 + .000023t$ $.1551 + .000023T$
Carbon dioxide	CO_2	$\begin{cases} .1877 + .000093t \\ .1421 + .000093T \end{cases}$	$.1423 + .000093t$ $.0967 + .000093T$
Water vapor	H_2O	$\begin{cases} .4228 + .000202t \\ .3234 + .000202T \end{cases}$	$.3117 + .000202t$ $.2123 + .000202T$
Nitrogen	N	$\begin{cases} .2429 + .000024t \\ .2312 + .000024T \end{cases}$	$.1714 + .000024t$ $.1587 + .000024T$
Oxygen	O	$\begin{cases} .2125 + .000021t \\ .2021 + .000021T \end{cases}$	$.1500 + 000021t$ $.1396 + .000021T$

If specific heat increases with temperature as rapidly as indicated by the results of Mallard and Le Chatelier, the thermal efficiency of the gas engines of to-day is already so high that very little improvement may be expected from anything we can do to perfect the thermal action of the cycle. If, on the other hand, the specific heat is constant within the limits of temperature found in gas-engine practice, there still remains a wide field for improvement in the direction of gas-engine efficiency.

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[1] Z. d. V. D. I., 1902, p. 1307.
[2] Z. d. V. D. I., 1903, p. 631.

The following table shows numerically the difference in the results, assuming one or the other of the above conditions, regarding specific heat. The experimental results were taken from the above mentioned "Untersuchungen" by Meyer.

TABLE 2

EFFECT OF VARIATION OF SPECIFIC HEAT

	Specific Heat Constant.		Specific Heat Variable.	
	Test No. 57	Test No. 64.	Test No. 57.	Test No. 64.
Temperature of combustion ....° F.	4746	3826	3439	2960
Temperature at end of adiabatic expansion ....° F.	2856	2285	2593	2153
Thermal efficiency of engine without losses (Ideal engine) %	39.75	40.30	29.69	31.78
Relative total work lost referred to the work delivered by Ideal engine .... %	33.46	34.50	10.95	16.97
Ratio between the work actually delivered to the possible amount of work that may be done by the working fluid between the volume limits given .... %	66.54	65.50	89.05	83.03

The values above given refer to a Deutz gas engine having a cylinder diameter of 8.67″, a stroke of 13″, and a ratio of compression = 3.84. In Test No. 57 the ratio of air to illuminating gas was 8.9 and the total load equal to 10.36 B.H.P. For Test No. 64 the corresponding figures were 12 and 6.13 B.H.P.

These figures show very clearly the importance of the question of the variability of specific heat with temperature in its bearing upon the theory of gas-engine design. A definite and final answer to this question is not to be expected for some time to come. To establish a working basis, therefore, for what is to follow, it is assumed, in accordance with the views of Zeuner, Fliegner, and others, that the specific heats c_p and c_v are constant within the ordinary ranges of temperature. The influence of *high pressures* upon the specific heat c_p has of late years been made the subject of scientific investigation, mainly by Prof. von Linde.[1] According to the results obtained by the latter investigator, atmospheric air shows the following relation:

$$c_p = c_{po}\left[1 + \frac{T^2}{\sqrt[3]{(T^3 - 3\alpha P)^2}}\right],$$

in which $c_{po} = .237$; P = pressure in kilograms per square meter; $\alpha = 20570$ = constant, and T is absolute degrees Centigrade.

The equation gives the following results:

For $P =$	10000	100000	400000	700000	Kg./sq. m.
$p =$	1	10	40	70	atmospheres
$=$	14.2	142	568	994	pounds/sq. in.
when $T = 273°$ C. $= 493°$ F. $c_p =$	.2375	.2419	.2510	.2779	
when $T = 373°$ C. $= 673°$ F. $c_p =$	.2372	.2389	.2448	.2511	

---

[1] Sitzungsberichte der Königlich Bayer, Akademie der Wissensch., 1897, No. 3.

According to this, the specific heat c_p is throughout smaller at 212° than at 32° (Mallard and Le Chatelier's formulæ would have shown the opposite result). The difference increases with an increase in the pressure. At any one temperature, however, c_p increases with the pressure, although in a decreasing ratio. For the two pressure extremes, $p=14.2$ lbs. per sq.in. and $p=994$ lbs. per sq.in., the increase of c_p at 32° is about 17%, while at 212° the increase is only 6%. Since in gas-engine practice the temperatures far exceed 1800° F., we may conclude that c_p for the fuel mixtures increases but very little with pressure. Hence the further assumption, for the time being, that the specific heats do not vary with pressure is justified.

Tables giving values of c_p and c_v for the perfect gases and fuel mixtures will be found in the Appendix.

3. Heating Value and Standard Condition of Fuel. In determining the efficiency of a gas engine, only the *lower* heating value of the fuel, i.e., the heat of combustion, after subtraction of the heat of vaporization of the water vapor carried by the gases of combustion, should be used. The reason for this is that the exhaust temperature always far exceeds the temperature of condensation of steam, the latter is discharged in vapor form and the heat of vaporization is never available as far as the engine is concerned.[1] The difference between lower and higher heating value (H_0) is of course proportional to the amount of hydrogen contained in the fuel, or to the amount of steam carried by the exhaust gases. In the case of illuminating gas, this difference may amount to 10% and is therefore quite considerable.

If one cubic foot or one pound of fuel of heating value H_u is mixed with L cu.ft. or lbs. of air, the heating value of the mixture will be

$$H_g=\frac{H_u}{1+L}\text{ B.T.U./cu.ft. (or B.T.U./lb.)}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (1)$$

It is usual to refer heating value and fuel consumption to a fixed set of **standard conditions.** Thus the fuel gases are usually referred to the mean atmospheric pressure of 29.92″ Hg., and a temperature of 32° F.[2] Thus, if for any given barometric pressure of b inches and a temperature of t or T degrees, the lower heating value of a fuel is H_u B.T.U., this value, by referring it to standard conditions will increase to

$$H'_u=H_u\frac{29.92}{b}\times\frac{461+t}{493}=\frac{29.92TH_u}{493b}\sim .061\frac{TH_u}{b}\text{ B.T.U./cu.ft.} \quad . \quad . \quad . \quad . \qquad (2)$$

On the other hand, any volume of gas v, measured at b'' Hg. and $t°$, by reference to standard conditions, decreases to

$$v_0=\frac{v}{1+.00203(t-32)}\times\frac{b}{29.92}=\frac{493v}{461+t}\times\frac{b}{29.92}=\frac{493vb}{29.92T}\sim 16.5\frac{vb}{T}. \quad . \quad . \quad . \quad . \qquad (3)$$

Fuel guarantees for illuminating-gas engines are usually based upon a heating value for the gas of 560 B.T.U. per cu.ft. under standard conditions. Suppose now that such an engine had consumed v cu.ft. of gas under b'' Hg. pressure and $t°$ temperature, and that the heating value of the gas was H_u B.T.U. per cu.ft. under these conditions. Then the gas volume reduced to standard conditions of pressure and temperature and to the standard of 560 B.T.U. per cu.ft., will be

---

[1] E. Meyer, Z. d. V. D. I., 1899, p. 283, in his thermodynamic investigations reaches the same conclusions regarding this much discussed point.

[2] For liquid fuel 59° or 60° F. is usually considered the standard reference temperature, but variations from this temperature have but a very slight effect upon the fuel consumption of oil engines.

$$v'_0=16.5\frac{vb}{T}\,\frac{H'_u}{560}.$$

But from eq. (2)
$$H'_u=\frac{.061TH_u}{b}.$$

hence
$$v'_0=16.5\frac{vb}{T}\times\frac{.061TH_u}{560b}=\frac{vH_u}{560}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (4)$$

It is clear from this that the opposite effects of b and t upon v or H_u cancel each other, and v'_0 therefore depends only upon $\frac{H_u}{560}$.

Example. $v=14.8$ cu.ft., $H_u=493$ B.T.U. per cu.ft. for $b=28.55''$ Hg. and $t=53°.6$, or $T=514°.6$ F.

Then
$$H'_u=.061\frac{514.6\times 493}{28.55}=540 \text{ B.T.U. per cu.ft.}$$

$$v_0=16.5\frac{14.8\times 28.55}{514.6}=13.5 \text{ cu.ft.}$$

Reducing to the standard heating value $=560$ B.T.U. per cu.ft., the volume will be

$$v'_0=13.5\frac{540}{560}=13.02 \text{ cu.ft.}$$

The standard conditions to which volumes are most commonly referred, i.e., 29.92″ Hg. and 32° F., are very badly adapted to gas-engine practice because, on this basis, all gas volumes are stated smaller, by from 8 to 15%, than the volumes actually used or indicated by the gas meter, which latter indication alone determines the fuel cost. Fuel consumption computed on the basis of equations (3) or (4) may therefore mislead gas-engine buyers or operators as to the real economy of the engines under consideration, which fact may in some cases have unpleasant consequences as far as the manufacturer is concerned.

It is of course not open to question that a fixed basis of comparison, i.e., a set of standard conditions, is fully as important to gas-engine manufacture and trade as it is for scientific investigation; but in order to arrive at such a basis, only such a pressure and temperature should be chosen, that the reduced data represent figures as closely as possible corresponding to actual results. This is most nearly the case when the **standard pressure** is taken as 28.92″ Hg. (=1 metric atmosphere), and the **standard temperature** that already commonly employed as such in chemistry and physics, i.e., 59° F. (=15° C.). In combination with these figures, the **standard heating value** for illuminating gas may well be placed at $H_u=560$ B.T.U./cu.ft.

With this proposed set of **standard conditions**, eq. (3) may be written in the following simple form:

$$v_0=\frac{520v}{461+t}\times\frac{b}{28.92}=\frac{17.977vb}{461+t}\sim\frac{18vb}{T}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (3a)$$

The approximation in the last form of the above equation causes an error of less than $\frac{2}{10}$ of 1% in the exact result.

Example. For $b=29.3''$ Hg., $t=71°.6$, and $v=31.7$ cu.ft, the value of v_0 is from

Eq. (3) Usual Standard Conditions:

$$v_0=16.5\frac{31.7\times 29.3}{532.6}=28.79 \text{ cu.ft.},$$

Difference $v-v_0=9.5\%$.

Eq. (3*a*) Proposed Standard Conditions:

$$v_0=18\frac{31.7\times 29.3}{532.6}=31.4 \text{ cu.ft.}$$

Difference $v-v_0=.95\%$.

From the above it would appear that a pressure of 28.92″ Hg. and a temperature of 59° F. represent the most suitable set of standard conditions for gas-engine practice.

4. Thermal Work and Efficiencies. In judging heat engines from the standpoint of economy, it is usual to compare the amount of heat converted into work with the amount furnished to the machine in the fuel. This assumes as an ideal case that every thermal unit furnished does an amount of work equivalent to 778 ft.-lbs., or that each pound of fuel of heating value H gives a return of

$$L_0=\frac{H}{A}=778H \text{ ft.-lbs.}$$

The duty or capacity of such an ideal engine, when the fuel consumption is C_s pounds per hour, would be

$$L_w=\frac{778HC_s}{33000\times 60}=\frac{HC_s}{2545} \text{ horse-power. . . . . } (5)$$

The indicated pressure diagram represents the real external work done. Comparing this diagram with the ideal pressure diagram of the cycle concerned furnishes a means of arriving at the magnitude and cause of the heat losses that occur.

Fig. 3, p. 11, shows in full line the ideal pressure diagram of an explosion engine, and in dotted line the real indicator diagram. The latter shows the indicated net work L_i, done by the amount of heat Q_1 furnished during one stroke. If the work shown by the ideal diagram is $\frac{Q}{A}=L_t$ ft.-lbs., there will be lost during every cycle an amount of heat equivalent to

$$L_-=L_t-L_i \text{ ft.-lbs.};$$

or

$$Q_-=AL_- \text{ B.T.U.}$$

This total loss L_- or Q_- is due to a series of causes, such as imperfect ignition and combustion, late ignition, and after-burning, conduction of heat during compression and expansion, etc., the relative influence of each of which can be judged only approximately from the indicator diagram. The ratio $\frac{L_i}{L_t}$ is called the *card factor*[1] of the pressure-volume diagram.

The area enclosed by the lines above and below the atmospheric pressure line depends for its size upon the suction and exhaust resistances, i.e., upon the design of the engine and not upon any internal thermal actions. It is therefore wrong, in thermal investigations, to subtract the lost work thus represented by the lower loop when determining the indicated thermal work L_i, as is often done, since the heat necessary to make up this loss must come from the heat in the fuel. Only when it is desired to determine the purely frictional loss of the machine, as compared with the total work done against internal resistance, is this method of computation justified. Otherwise the subtraction of the fluid loss (lower loop loss) would make the cyclic efficiency (η_i) less and the mechanical efficiency (η_m) of the machine greater than they really are.

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[1] In German "Völligkeitsgrad," an expression practically untranslatable in a single word. The expression "card factor" is much used in steam-engine practice to represent what is practically the same thing.

If the mean indicated pressure of the indicator diagram is p_i pounds per square inch, or P_i pounds per square foot, and if the piston displacement is V_h cu.ft., the indicated work of one stroke will be

$$L_i=144p_iV_h=P_iV_h \text{ ft.-lbs.} \quad \ldots\ldots\ldots \quad (7)$$

The corresponding thermal equivalent is

$$Q_i=AL_i=\frac{144p_iV_h}{778}=\frac{P_iV_h}{778} \text{ B.T.U.}$$

$$Q_i=.185p_iV_h=.001284P_iV_h \text{ B.T.U.} \quad \ldots\ldots\ldots \quad (8)$$

Assuming that the capacity of the engine is to be N_i I.H.P. at n r.p.m., the piston displacement will have to be

$$V_h=\frac{33000\times N_i\times 2}{144p_in}=\frac{458.06N_i}{np_i}=\frac{66000N_i}{nP_i} \text{ cu.ft.} \quad \ldots\ldots \quad (9)$$

for the 4-cycle engine, and

$$V_h=\frac{33000\times N_i}{144p_in}=\frac{229.03N_i}{np_i}=\frac{33000N_i}{nP_i} \text{ cu.ft.} \quad \ldots\ldots \quad (10)$$

for the 2-cycle machine.

If an engine uses C_s pounds or cu.ft. of fuel per hour of heating value $=H$ B.T.U. per pound or cu.ft., the amount of heat actually expended per combustion stroke is

$$Q=\frac{C_sH}{\frac{60}{2}n} \text{ B.T.U. in a single-acting 4-cycle,} \quad \ldots\ldots \quad (11)$$

and $$Q_1=\frac{C_sH}{60n} \text{ B.T.U. in a single-acting 2-cycle engine.} \quad \ldots \quad (12)$$

The work equivalents of these amounts of heat are

$$L=\frac{Q}{A}=\frac{778C_sH}{30n}=\frac{25.93C_sH}{n} \text{ ft.-lbs. for the 4-cycle,} \quad \ldots\ldots \quad (13)$$

and $$L=\frac{778C_sH}{60n}=\frac{12.96C_sH}{n} \text{ ft.-lbs. for the 2-cycle.} \quad \ldots\ldots \quad (14)$$

From the above equations the **indicated thermal efficiency**, in short, the *indicated efficiency* of the cycle (see p. 11) is

$$\eta_i=\frac{AL_i}{Q}=\frac{Q_i}{Q}=\frac{N_i\times 33000\times 60}{778HC_s}$$

$$=\frac{\mathbf{2545N_i}}{\mathbf{HC_s}} \quad \ldots\ldots\ldots\ldots \quad (15)$$

Dividing the indicated thermal efficiency η_i by the *theoretical thermal efficiency* η_t of the cycle, we obtain a ratio which expresses the relation between the real machine and another machine which is assumed to have no losses. This ratio is again analogous to the *card factor*, which may then be expressed by

$$\eta_g=\frac{\eta_i}{\eta_t}, \qquad \text{(16)}$$

or using the expressions above developed, for the work equivalents,

$$\eta_g=\frac{L_i}{L_t}. \qquad \text{(16}a\text{)}$$

A certain part of the indicated work N_i developed by the machine is lost on the way from cylinder to shaft through friction of the machine. If we assume that this negative (friction) work is equal to N_r expressed in I.H.P., the useful (effective) engine horse-power will be

$$N_e=N_i-N_r. \qquad \text{(17)}$$

The mechanical efficiency of the machine will then be

$$\eta_m=\frac{N_i-N_r}{N_i}=\frac{N_e}{N_i}. \qquad \text{(18)}$$

Substituting for N_i, N_e and N_r, the corresponding mean pressures $p_i=p_e+p_r$ pounds per square inch, we may also write

$$\eta_m=\frac{p_i-p_r}{p_i}=\frac{p_e}{p_i}. \qquad \text{(18}a\text{)}$$

Finally the product of the three efficiencies above developed gives the **economic efficiency**

$$\eta_w=\eta_g\eta_t\eta_m=\eta_i\eta_m. \qquad \text{(19)}$$

The economic efficiency referred to the basis of heat consumption may also be expressed by

$$\eta_w=\frac{N_i\eta_m\times 33000\times 60}{778HC_s}=\frac{2545N_e}{HC_s}. \qquad \text{(19}a\text{)}$$

If instead of using the fuel consumption C_s per hour, the consumption C per horse-power per hour is introduced, we will have the general equation

$$\eta_w=\frac{\mathbf{2545}}{\mathbf{HC}}. \qquad \text{(20)}$$

As determined by experience, η_g varies from .40 to .75 according to size and quality of engine. $\eta_m=.75$ to .90. Values for η_t are given in Tables 3 and 4, p. 13, and 19.

Example.
$$\left.\begin{array}{l} \eta_t \quad\; \eta_g \quad\; \eta_m \quad\; \eta_w \\ \underbrace{.44\times .72}_{\eta_i=.317}\times .90=.285 \end{array}\right\} \text{Energy loss}=1-.285=.715=71.5\% \text{ of the heat furnished.}$$

The relation that the various heat losses bear to one another can only be determined approximately on account of the uncertainty prevailing with respect to the variation of specific heats and dissociation temperatures. It is evident, however, that the two main sources of loss are found in the cooling water and in the exhaust gases.

The inter-relation of the heat converted into work and the heat losses is best made clear by means of the *entropy diagram* in which the ordinates represent absolute temperatures (T), the abscissæ the quotient between quantity of heat and absolute temperature ($Q \div T$, see under Entropy in Appendix), and areas represent quantities of heat (Q). The entropy diagram, Fig. 2, has been derived from the indicator diagram, Fig. 1, which was obtained from one of the later models of the 8 H.P. Körting illuminating-gas engine.[1] The broken lines of Fig. 2 represent the ideal heat diagram that would have been obtained with a machine operating without loss, assuming constant specific heat. (Regarding the construction and evaluation of such diagrams, see Fig. 23, p. 41). The real diagram, shown by the hatched area, Fig. 2, represents only about $\frac{1}{3}$ of the area of the ideal diagram ($\eta_i=33.1\%$, to be exact). Area $V_1=34.9\%$ represents the heat loss to the cooling water, area $V_2=33\%$, the heat lost in the exhaust gases.

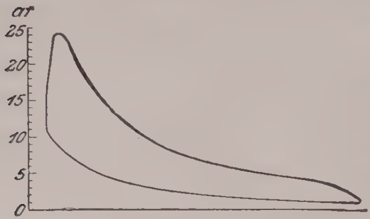

FIG. 1.

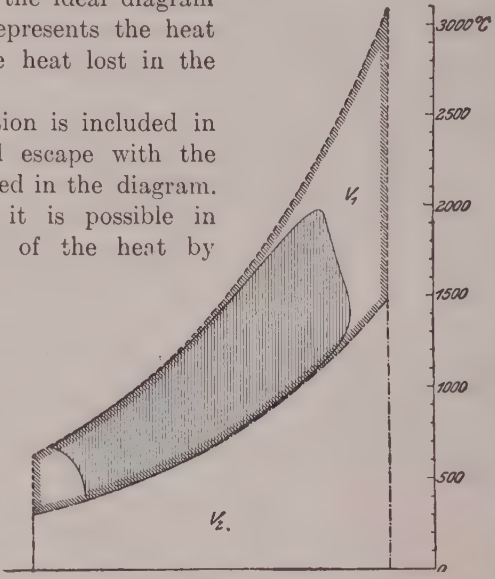

FIG. 2.

The heat lost due to incomplete combustion is included in area V_2. Since the gas particles not burned escape with the exhaust gases, this loss by itself is not represented in the diagram. But since, as experience has demonstrated, it is possible in very unfavorable cases to lose from 10–20% of the heat by incomplete combustion, this source of loss is by no means negligible. This is of special importance since it furnishes the designer with a valuable means of improving our present-day engines.

(See also Part II, Valves, and Part IV, Combustion.)

E. Meyer concludes his "Untersuchungen am Gasmotor," quoted on p. 4, with the following words: "It cannot be denied that, in many engines in operation to-day, especially those employing the leaner gases, large quantities of fuel gas pass through the engine unburned, and that the economy of many engines can be improved mainly by counteracting the heat losses due to incomplete combustion as far as possible by a careful mixing of the new charge." He bases his statement mainly on some of his experimental data given in a table in Part IV, Subdivision III, of this book.

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[1] From a lecture by C. Linde on "die Auswertung der Brennstoffe als Energieträger," given before the Verein d. I., 1903. Z. d. V. D. I., 1903, p. 1509.

B. THE VARIOUS CYCLES OF OPERATION

I. The Constant-volume Cycle

1. The thermodynamic examination of the various working cycles is based on the assumption, not quite true in actual practice, that the charge, as the carrier of heat, is a perfect, ideal gas, which, in a non-conducting cylinder, passes through a reversible cycle of operations. With this assumption, the resulting pressure diagram is an area bounded by four lines, along one of which heat is furnished, along another heat is abstracted, while the other two are adiabatics.

If, in such a cycle, Q_1 B.T.U. are furnished to the gas, while Q_2 B.T.U. are rejected, so that $Q_1-Q_2=Q$ B.T.U. are transformed into external work, then in general the thermal efficiency of the cycle is (see p. 8 and Appendix).

$$\eta'_t=\frac{Q_1-Q_2}{Q_1}=1-\frac{Q_2}{Q_1}=\frac{Q}{Q_1}. \quad . \quad . \quad (21)$$

Now in the constant-volume cycle, whose theoretical pressure diagram is shown in Fig. 3, the heat furnished during explosion,

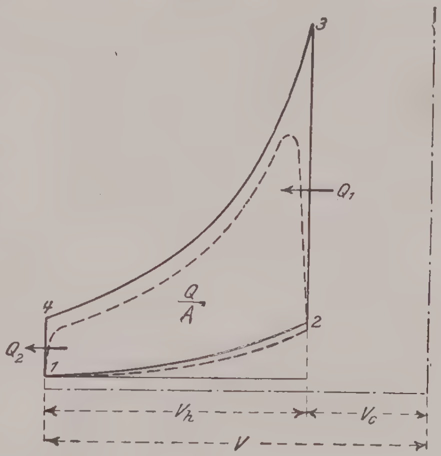

FIG. 3.

$$Q_1=Gc_v(T_3-T_2) \text{ B.T.U.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (22)$$

the heat in the exhaust gases,

$$Q_2=Gc_v(T_4-T_1) \text{ B.T.U.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (23)$$

hence

$$Q=Q_1-Q_2=Gc_v(T_3-T_2-T_4+T_1) \text{ B.T.U.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (24)$$

Eq. (24) represents the external work done. G is the weight of the charge consisting of G_r pounds of burned gases remaining in the cylinder from the previous combustion, G_b pounds of fuel, and G_l pounds of air. Further, c_v is the specific heat at constant volume, while T and P represent absolute temperatures and pressures for points on the diagram marked with the corresponding subscripts.

Eq. (24) may be written

$$Q=Gc_v\left[T_3\left(1-\frac{T_4}{T_3}\right)-T_2\left(1-\frac{T_1}{T_2}\right)\right]. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (24a)$$

From this, by use of Poisson's Law, according to which $\frac{T_2}{T_3}=\frac{T_1}{T_4}$, and by substituting the value of Q_1, eq. (22) for the left half of the expression in parenthesis in eq. (24) we finally derive that

$$Q=Q_1\left(1-\frac{T_1}{T_2}\right). \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (24b)$$

Hence eq. (21) becomes

$$\eta_t = \frac{Q_1\left(1 - \frac{T_1}{T_2}\right)}{Q_1} = 1 - \frac{T_1}{T_2}.$$

But from the same law,

$$\frac{T_1}{T_2} = \left(\frac{V_c}{V}\right)^{x-1} = \left(\frac{P_1}{P_2}\right)^{\frac{x-1}{x}},$$

therefore we may also write

$$\eta_t = 1 - \left(\frac{V_c}{V}\right)^{x-1} = 1 - \left(\frac{P_1}{P_2}\right)^{\frac{x-1}{x}}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (25)$$

In eq. (25) $\frac{V_c}{V}$ is the reciprocal of ε, the ratio of compression (which in the case of this cycle may be taken equal to the ratio of expansion δ). Hence finally

$$\eta_t = \mathbf{1} - \frac{1}{\varepsilon^{x-1}} = \mathbf{1} - \varepsilon^{1-x}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (25a)$$

The same final result is naturally obtained when the derivation of the value for the efficiency is based upon the energy equivalent L instead of upon the quantity of heat Q. In this case it will simplify matters if the computation is made with the specific volume $v = \frac{V}{G}$ of one pound of the charge.

The absolute energy represented by the expansion stroke is

$$L_a = \frac{P_3 v_c}{x-1}\left[1 - \left(\frac{v_c}{v}\right)^{x-1}\right] = \frac{P_3 v_c}{x-1}\left(1 - \frac{1}{\varepsilon^{x-1}}\right). \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (26)$$

The work of the compression stroke is

$$L_c = \frac{P_2 v_c}{x-1}\left[1 - \left(\frac{v_c}{v}\right)^{x-1}\right] = \frac{P_2 v_c}{x-1}\left(1 - \frac{1}{\varepsilon^{x-1}}\right). \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (26a)$$

Subtracting (26a) from (26), the available energy of the theoretical diagram will be

$$L_t = \frac{P_3 v_c}{x-1}\left(1 - \frac{1}{\varepsilon^{x-1}}\right) - \frac{P_2 v_c}{x-1}\left(1 - \frac{1}{\varepsilon^{x-1}}\right)$$

$$= \frac{P_3 v_c - P_2 v_c}{x-1}\left(1 - \frac{1}{\varepsilon^{x-1}}\right) = \frac{(P_3 - P_2) v_c}{x-1}\left(1 - \frac{1}{\varepsilon^{x-1}}\right). \quad . \quad . \quad . \quad . \quad . \quad (27)$$

Now the energy equivalent of the amount of heat Q furnished the cycle is

$$L = \frac{Q_1}{A} = \frac{c_v v_c}{AR}(P_3 - P_2) = \frac{(P_3 - P_2) v_c}{x-1}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (28)$$

From (27) and (28) then

$$\frac{L_t}{L} = \eta_t = \frac{\frac{(P_3 - P_2) v_c}{x-1}\left(1 - \frac{1}{\varepsilon^{x-1}}\right)}{\frac{(P_3 - P_2) v_c}{x-1}}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (29)$$

or, as before,

$$\eta_t = 1 - \frac{1}{\varepsilon^{x-1}}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (29a)$$

Eqs. (25) *and* (25*a*) *show that the thermal efficiency of the constant-volume engine depends first upon the ratio of compression* ε, *increasing as the compression pressure* P_2 *increases, and second, that it varies in direct ratio with the value of* $x=\frac{c_p}{c_v}$. It is true that the influence of x is small as compared with that of a variation in ε, but the difference may amount to 6 or 8% of the minimum value, as shown by the following

Example. Assume the following constants for two different illuminating-gas mixtures:

Mixture.	Ratio Air to Gas.	ε	x	
I	1 to 6	5	1.354,	then $\eta_t=1-(\frac{1}{5})^{354}=.435$
II	1 to 13.5	5	1.383,	then $\eta_t=1-(\frac{1}{5})^{.383}=.461$

$$\text{Difference}=\frac{.461-.435}{.435}=.06=6\%.$$

The weaker mixture therefore shows a thermal efficiency 6% higher than that of the rich mixture. An additional advantage is that in practical operation, the weaker mixture burning at a lower temperature will lose less heat to the jacket water. It is therefore advisable in gas-engine practice to use lean mixtures and to compress these to a high degree.[1]

TABLE 3

THERMAL EFFICIENCIES η_t OF THE CONSTANT-VOLUME CYCLE FOR VARIOUS VALUES ε AND x

For $\varepsilon=$	2.0	2.5	3.0	3.5	4.0	4.5	5.0	6.0	7.0	8.0	9.0	10
For $x=1.20$	0.129	0.167	0.197	0.221	0.242	0.260	0.275	0.301	0.322	0.340	0.356	0.369
For $x=1.25$	0.159	0.205	0.240	0.269	0.293	0.313	0.331	0.361	0.385	0.405	0.423	0.438
For $x=1.30$	0.188	0.241	0.281	0.313	0.340	0.363	0.383	0.416	0.442	0.464	0.483	0.499
For $x=1.35$	0.216	0.274	0.319	0.355	0.384	0.409	0.431	0.466	0.494	0.517	0.537	0.553
For $x=1.40$	0.248	0.313	0.363	0.402	0.434	0.460	0.483	0.520	0.550	0.574	0.594	0.611

2. Practical Considerations. The rule just developed does not of course apply in practice without qualification. In the first place, contrary to the assumptions above made, the real cycle is not a closed one. Further, the expansion and compression lines cannot be adiabatic, because the cycle is carried out in a cylinder the walls of which are strongly cooled; and lastly, the charge does not behave like a perfect gas because chemical changes occur during the cycle. Nevertheless, its accuracy is attested by the fact that in practice the indicated efficiency η_i can be considerably improved by increasing the amount of compression. In order to produce this net result, therefore, it must be assumed that the contradictory influences at work during the cycle operate in such manner that those tending to increase the efficiency balance, or rather overbalance, those tending to lower it. The purely practical advantages of high compression are in themselves of some importance; it increases the temperature of the charge before ignition, decreases the distances through which the flame must propagate, decreases the external cooling surfaces, and diminishes the proportional admixture of the burned gases to the new charge. All of these effects tend to improve the process of combustion and to decrease the loss to the cooling water. In the endeavor to utilize to its fullest degree this means of improving the economy of internal-combustion engines, the compression pressure has been increased step by step from 3 or 4 atm. to 8 or 10 (in the case of illuminating gas) and to 12 or 16 atm. (in the case of lean gases). Attempts are made to raise the upper limits even beyond these values, but it must be remembered that the upper limit is set by the approach of the compression temperature to the ignition temperature of the charge. It is therefore interesting to examine into the degree of success these endeavors are likely to have.

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[1] The advantages of the use of lean mixtures, as regards the effect on thermal efficiency, are especially marked if it is assumed that specific heats increase with temperatures. See p. 4.

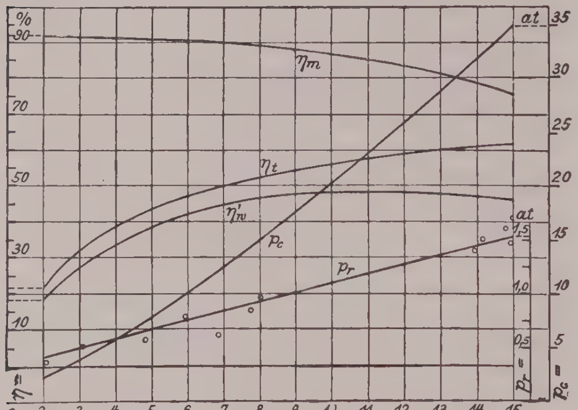

Fig. 4.—Relation between Compression and Efficiency.

Fig. 4 shows the increase of η_t with a variation in the compression ratio ε from 2 to 15, assuming that x is constant at 1.35. It will be noted that the increase in η_t is less and less rapid, and that beyond $\varepsilon = 15$ it increases but very little. But, as has been shown on p. 9, the practical criterion of the value of a machine is not the value of η_t, but that of $\eta_w = \eta_g \eta_t \eta_m$. The effect of variation in the compression upon the factor η_g is indirect and is difficult to express numerically. In any case it is small, and the injurious effect, for instance, that a lowering of the compression may have upon the process of combustion may be easily counteracted by making the charge richer and thereby obtaining more effective ignition, etc. If therefore η_g is neglected, we may write, for all degrees of compression

$$\eta'_w = \eta_t \eta_m, \quad \ldots \ldots \ldots \quad (30)$$

which states that the economy of the gas engine is a function of its thermal and mechanical efficiencies only. But while η_t may be accurately computed in all cases, we possess absolutely no numerical basis for the determination of η_m. There is up to to-day no formula which gives the frictional resistance likely to be encountered in internal-combustion engines with even a fair degree of accuracy. We are therefore compelled to depend upon experience, which so far seems to have brought out the following facts:

(*a*) That the mean frictional resistance p_r, based on one square inch of piston area, increases with the compression, all other conditions remaining the same; and further

(*b*) That this frictional resistance p_r is either entirely independent of the mean indicated pressure p_i or varies with it in the *inverse ratio*. (Compare for instance the experimental data in several tables in Part III.)

To obtain a fairly approximate idea of the value of p_r, a curve has been drawn in Fig. 4, showing the relation between p_r and ε for a number of the larger gas engines. An examination of the available data shows that, excluding some very irregular figures, p_r varies according to a law that may be approximately expressed by

$$p_r = .94\sqrt{p_c} \text{ lbs. per sq.in.} \quad \ldots \ldots \ldots \quad (31)$$

The derivation of an absolutely general exact equation fails on account of the varying conditions of design and of operation. The relative positions of the p_c and p_r

curves in Fig. 4 indicate that p_r increases at a much slower rate than p_c. Now from eq. (30), η'_w does not depend upon p_r but upon

$$\eta_m = \frac{N_e}{N_i} = \frac{p_e}{p_i}$$

and since $p_e = p_i - p_r$, that is, since

$$\eta_m = \frac{p_i - p_r}{p_i}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (32)$$

it follows that an increase in the mean frictional resistance of the engine does not decrease the mechanical efficiency η_m, provided there is an adequate increase in the specific capacity of the engine, as measured by the factor p_i. To this condition is principally due the fact that our modern engines, working with compression pressures from two to four times those employed in the first Otto engines, still show a mechanical efficiency fully the equal of that found in these early machines.

Eq. (30) in conjunction with eq. (25*a*) gives

$$\eta'_w = \eta_t \frac{p_i - p_r}{p_i} = \left(1 - \frac{1}{\varepsilon^{x-1}}\right)\left(\frac{p_i - p_r}{p_i}\right), \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (30a)$$

from which it follows that *a further increase of compression ceases to be beneficial as soon as it is obtained at the expense of* p_i, *i.e., as soon as it is rendered possible only by making the mixtures extremely lean.*

The curve for η_m in Fig. 4 was constructed by computing values for the mechanical efficiency from eq. (32), substituting values of p_i as found from practice. This curve shows that up to $\varepsilon = 6$, i.e., $p_c =$ appr. 140 lbs. per sq.in., the decrease of η_m is but slight, but after that its value drops rapidly. The reason for this decrease is twofold; on the one hand it is due to the increase of all the engine dimensions to resist the increasing pressures (see p. 68, Part II); on the other, it is probably largely due to the fact that the mixtures will have to depart more and more from the normally best mixture as the compression temperatures approach that of pre-ignition of the charge. The use of leaner mixtures results in lower values of p_i, and this, according to eq. (32), means a decrease in η_m.

The product $\eta_t \eta_m$ is finally shown in the curve marked η'_w, Fig. 4. Up to $\varepsilon = 8$, for which $p_c =$ appr. 210 lbs. per sq.in., the rate of increase of η'_w is quite rapid. The maximum value is reached for $\varepsilon = 10$, $p_c =$ appr. 280 lbs. per sq.in. Beyond this, an increase of ε, instead of showing a further increase, shows a decrease in η'_w. *We conclude from this that in constant-volume engines the use of compression ratios beyond* $\varepsilon = 8$ *does not result in further material gains in the economic efficiency, and that the economic limit of maximum compression will lie between* 210 *and* 280 *lbs. per sq.in.* This conclusion does not apply to constant-pressure engines, because for these the value of η_t follows a different law, and because η_m decreases more slowly on account of the generally greater value of p_i (see p. 9).

The probable maximum value of the *explosion pressure* p_z may be found from the following considerations: from p. 34, we have

$$\frac{p_z}{p_c} = \frac{T_z}{T_c}, \quad \text{and} \quad T_z = T_c + \frac{Q_1}{c_v G_l},$$

i.e., the increase in pressure is proportional to the temperature increase, and the latter is proportional to Q_1. High explosion pressures, desirable from the standpoint of high specific engine capacity, can therefore be obtained only together with high temperatures, which means with rich mixtures. But such mixtures cause greater heat losses in cooling water and in exhaust, to say nothing of some practical difficulties. Engines using rich illuminating-gas mixtures with comparatively low compression pressures show an average pressure ratio in the neighborhood of $\frac{p_z}{p_c}=4-5$. The same mixtures used with high compressions would produce explosion pressures exceeding 50 atm. and explosion temperatures far in excess of 3800° F. To avoid such dangerous pressures and temperatures, the means generally used is to decrease the heat contents of the charge in engines using high compressions, so that the explosion pressure shall be from 2.5 to 3 times the pressure of compression, i.e., from 25 to 30 atm. Under such conditions the dimensions of the engine parts remain within reasonable limits, and the heat is much better utilized, as experience has shown. It is quite likely, however, that in time the economic compression limit of from 16–18 atm., which was referred to above, will be approached, resulting in explosion pressures in the neighborhood of 40 atm.=app. 575 lbs. The use of the still higher working pressures cannot be justified either on the ground of economy or from the standpoint of design.

II. The Constant-pressure Cycle

1. Thermodynamic Examination of the Constant-pressure Cycle. In the older constant-pressure engines (Brayton, Hargreaves, Brünnler, etc.) the taking in and compression of the charge, combustion, expansion, and exhaust were carried on in two separate cylinders. By combining in Fig. 5 the pump diagram $\overline{a\,1\,2\,b}$ with the power diagram $\overline{o\,a\,b\,3}$, we obtain the net energy diagram $\overline{0\,1\,2\,3}$. used in the investigation.

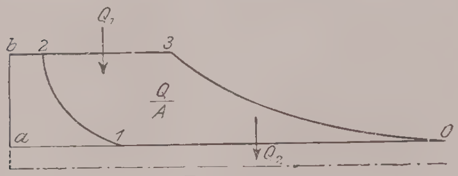

FIG. 5.

If it is assumed that G pounds of burned gases are expanded to *atmospheric* pressure p_0 lbs. per sq.in., we have, see Fig. 5,

Heat supplied, $$Q_1=Gc_p(T_3-T_2)\text{ B.T.U.} \quad (33)$$

Heat discharged, $$Q_2=Gc_p(T_0-T_1)\text{ B.T.U.} \quad (34)$$

Hence heat transformed is

$$Q=Q_1-Q_2=Gc_p(T_3-T_2-T_0+T_1)\text{ B.T.U.} \quad (35)$$

and the thermal efficiency

$$\eta_t=\frac{Gc_p(T_3-T_2-T_0+T_1)}{Gc_p(T_3-T_2)}=1-\frac{T_0-T_1}{T_3-T_2}. \quad (36)$$

Assuming adiabatic compression and expansion,

$$\frac{T_0}{T_3}=\frac{T_1}{T_2}, \quad \text{or} \quad \frac{T_0}{T_1}=\frac{T_3}{T_2}.$$

Hence $$\eta_t=1-\frac{T_0}{T_3}, \quad \text{or} \quad \eta_t=1-\frac{T_1}{T_2}. \quad (36a)$$

According to Poisson's law, however,

$$\frac{T_2}{T_1}=\left(\frac{V}{V_c}\right)^{x-1}=\varepsilon^{x-1}$$

and therefore finally

$$\eta_t=1-\frac{T_1}{T_1\varepsilon^{x-1}}=1-\frac{1}{\varepsilon^{x-1}}=1-\varepsilon^{1-x} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (36b)$$

The expression for the thermal efficiency of this cycle is therefore the same as that of the constant-volume cycle, eq. (25a) showing that the efficiencies of the two cycles are the same when the ratio of compression is the same in both. It should be observed, however, that for the same cylinder volume or piston displacement in each case, the constant-pressure cycle is less efficient than the constant-volume cycle because of the necessarily lower compression pressure. If, on the other hand, the same maximum combustion pressure be assumed in each case the balance is in favor of the constant-pressure cycle. On account of the resulting high compression temperatures in the constant-pressure cycle, the latter can then only be carried out by separate introduction of fuel. This is the principle of the *Diesel engine,* to which Fig. 6 and the following computations mainly refer. The final result of the investigation is not affected in the least if the ignition of the charge, instead of being brought about by the heat of compression, were caused by any other means.

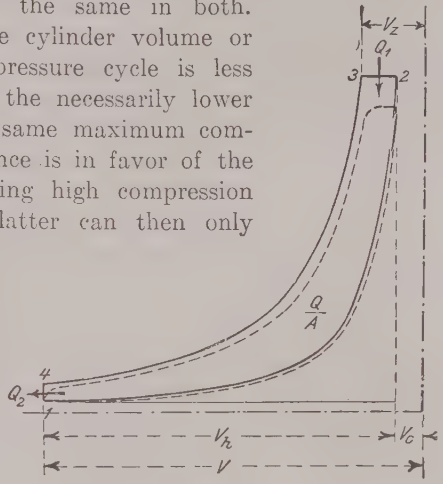

FIG. 6.

In constant-pressure engines already constructed, the compression ratio $\varepsilon=\frac{V}{V_c}$ is generally greater than the expansion ratio $\delta=\frac{V}{V_z}$. The cut-off ratio $\rho=\frac{V_z}{V_c}$ varies with the load on the engine. This fact affects the efficiency of the process, as shown below.

With the notation of Fig. 6,

$$Q_1=Gc_p(T_3-T_2), \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (37)$$

$$Q_2=Gc_v(T_4-T_1), \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (38)$$

and the heat transformed to external work again is

$$Q=Gc_p(T_3-T_2)-Gc_v(T_4-T_1)=Gc_p\left(T_3-T_2-\frac{T_4-T_1}{x}\right) \quad . \quad . \quad . \quad . \quad (39)$$

Now $T_3=T_2\frac{V_z}{V_c}=T_2\rho$, and $T_4=T_1\left(\frac{V_z}{V_c}\right)^x=T_1\rho^x$.

Hence $$Q_1=Gc_pT_2(\rho-1). \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (37a)$$

and $$Q_2=Gc_vT_1(\rho^x-1), \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (38a)$$

from which $$\eta_t=1-\frac{Q_2}{Q_1}=1-\frac{Gc_vT_1(\rho^x-1)}{Gc_pT_2(\rho-1)} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (40)$$

But $\frac{c_v}{c_p}=\frac{1}{x}$, and $\frac{T_1}{T_2}=\frac{1}{\varepsilon^{x-1}}$, hence finally

$$\eta_t=1-\frac{1}{x}\,\frac{\rho^x-1}{\varepsilon^{x-1}(\rho-1)} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (40a)$$

Still clearer is the following derivation of the final equation:

$$\eta_t = 1 - \frac{Gc_v(T_4 - T_1)}{Gc_p(T_3 - T_2)} = 1 - \frac{T_1}{xT_2}\left(\frac{\frac{T_4}{T_1} - 1}{\frac{T_3}{T_2} - 1}\right), \quad \ldots \ldots \quad (41)$$

or, since for the constant-pressure line we can write

$$\frac{T_3}{T_2} = \frac{V_z}{V_c}; \quad \text{and} \quad \frac{T_1}{T_2} = \left(\frac{V_c}{V}\right)^{x-1}; \quad \text{and} \quad \frac{T_4}{T_3} = \left(\frac{V_z}{V}\right)^{x-1},$$

we obtain

$$\eta_t = 1 - \frac{T_1}{xT_2}\left[\frac{\left(\frac{T_3}{T_2}\right)^x - 1}{\frac{T_3}{T_2} - 1}\right] = 1 - \frac{T_1}{xT_2}\left[\frac{\left(\frac{V_z}{V_c}\right)^x - 1}{\frac{V_z}{V_c} - 1}\right], \quad \ldots \ldots \quad (41a)$$

which, by substituting $\frac{T_1}{T_2} = \frac{1}{\varepsilon^{x-1}}$, and $\frac{V_z}{V_c} = \rho$, reduces to the form of eq. (40a).

The same equation may be derived by using the ratio of the energy equivalent L of the heat Q_1 supplied to the work L_t represented by the theoretical pressure diagram, Fig. 6, as follows:

Basing the computation again on the specific volume $v = \frac{V}{G}$ of one pound of the charge, we have

$$L = \frac{Q_1}{A} = \frac{c_p P_2}{AR}(v_z - v_c) = \frac{x}{x-1}P_2(v_z - v_c) \quad \ldots \ldots \ldots \quad (42)$$

The total energy of the combustion and expansion stroke is

$$L_a = P_2(v_z - v_c) + \frac{P_3 v_z}{x-1}\left(1 - \frac{1}{\delta^{x-1}}\right). \quad \ldots \ldots \ldots \quad (43)$$

The work of compression is found from

$$L_c = \frac{P_2 v_c}{x-1}\left(1 - \frac{1}{\varepsilon^{x-1}}\right). \quad \ldots \ldots \ldots \quad (44)$$

The available energy then is

$$L_t = \frac{Q_1 - Q_2}{A} = \left[P_2(v_z - v_c) + \frac{P_3 v_z}{x-1}\left(1 - \frac{1}{\delta^{x-1}}\right)\right] - \frac{P_2 v_c}{x-1}\left(1 - \frac{1}{\varepsilon^{x-1}}\right), \quad \ldots \quad (45)$$

from which, by dividing by the value of L from eq. (42), it finally follows again that

$$\eta_t = \frac{L_t}{L} = 1 - \frac{1}{\varepsilon^{x-1}}\left[\frac{\rho^x - 1}{x(\rho - 1)}\right].$$

Therefore the thermal efficiency of the constant-pressure engine (of the Diesel type), besides depending upon the values of the compression ratio ε and the ratio of specific heats x, also depends upon the cut-off ratio ρ. Further, the influence of this factor is very marked, as may be seen from Table 4, which has been computed from eq. (40a), assuming that $p_2 = \frac{P_2}{144} = 467$ lbs., $x = 1.30$ and 1.41, and $\rho =$ from 1.50 to 3.0.

TABLE 4

THERMAL EFFICIENCIES (η_t) OF THE CONSTANT-PRESSURE CYCLE FOR $P_2=467$ LBS. AND VARYING VALUES OF ρ, ε, AND x

	$\rho=$	1.50	1.75	2.0	2.25	2.50	2.75	3.0
$x=1.3$	$\varepsilon=16$	$\eta_t=0.535$	0.522	0.512	0.499	0.488	0.479	0.471
$x=1.41$	$\varepsilon=13$	$\eta_t=0.616$	0.602	0.588	0.576	0.564	0.552	0.540

The inter-relation of the various factors expressed by eq. (40*a*), i.e., the rule for the value of η_t, finds full corroboration in practice in that the indicated thermal efficiency η_i of Diesel engines always shows an appreciable increase for decreasing loads, that is, for smaller values of ρ and greater values of x.

In constant-pressure engines as constructed, owing to the injection of the fuel, there is an increase in the charge weight G during combustion. In the case of liquid fuels, however, this increase is not considerable and in any case, on account of the variation in amount, it would be very difficult to consider its effect in a formula generally applicable. Eqs. (37) to (45) therefore do not apply to oil engines with absolute rigidity. The exact treatment of the case would give slightly lower values for η_t, because, owing to the increase in G, the heat discharged Q_2 will be somewhat greater. In the case of constant-pressure engines using gas, this increase in charge weight, however, cannot be neglected in thermal investigations, because, depending upon the kind of gas, the fuel might under certain conditions constitute one-half of the total charge. It is unnecessary to take this matter up in detail at this time since constant-pressure gas engines (of the Diesel type) are to-day of no practical importance.

2. Practical Considerations. The Diesel constant-pressure engine for liquid fuels (the only representative of this type of engine) at rated load usually works with a cut-off volume equal to 10%, for which $\rho=2.5$. Assuming that $x=1.41$, the thermal efficiency of this cycle then is $\eta_t=.564$, from Table 4. A constant-volume engine showing the same maximum explosion pressure (=approximately 467 lbs.) would have to show a compression pressure of from 185 to 210 lbs. per sq.in., that is $\varepsilon=7$. Assuming again that $x=1.41$, the thermal efficiency of this cycle from eq. (25) or (25*a*) would be $\eta_t=.55$. The difference is therefore only 1.5% in favor of the constant-pressure cycle. But this advantage disappears with greater values of ρ, and with $\rho=2.75$ the balance is already the other way. Based upon the conditions of equal maximum pressures in the cycle, which is the only correct basis from the standpoint of both design and practice, the two methods of operation are therefore very nearly equal. The constant-pressure cycle offers a fundamental advantage only then when its maximum pressures of operation far exceed the limit of maximum pressures, set by ignition, for the constant-volume cycle. That, however, is not the case in the present Diesel engines, because the maximum pressures of from 475–500 lbs. occurring in these engines can be easily reached and safely handled in constant-volume engines. The undoubted superiority shown by Diesel engines is therefore not so much due to the different cycle of operation used by them, but mainly to some purely practical features, among which may be mentioned very perfect vaporization of the fuel oil and the instantaneous mixing and combustion of the oil vapor. These advantages, however, are peculiar to the oil engine only, and would have no bearing if gas were used as fuel. For gas fuel, therefore, the constant-pressure principle possesses theoretically no advantage over constant-volume operation; in practice, from the view-point of design and operation, the former even labors under a decided disadvantage as compared with the latter.

III. Thermodynamic Aspects of the Two Methods of Operation

The appearance of Diesel's pamphlet and, a little later, of his engine, was the direct cause of a number of scientific investigations on the various methods of operation which did much to clear the old question concerning the *best* operating cycle.

To supplement the author's own view of the question, as outlined in the previous paragraphs, the essentials of the most important of the investigations mentioned are given below.

1. Prof. Schöttler, in his thermodynamic examination of the Diesel cycle, comes to the following conclusion:[1] "Compared to the constant-volume cycle, the efficiency of the Diesel cycle is very inferior, unless very lean mixtures and consequently large cylinders are used. The latter cycle, however, possesses the advantage of lower temperatures and also obtains the benefit of the good effect, already mentioned, that lean mixtures have on efficiency. In fact, to the use of lean mixtures may be traced the excellent regulation of Diesel engines."

Schöttler carried out a very extended investigation on the three methods of operation (combustion at constant volume, at constant pressure, and at constant temperature). Some of the results of his numerical comparisons are given in Table 5. In summing up his final conclusion he says: "From all this it follows that the cycle introduced by Diesel certainly has undoubted advantages, but that it is quite possible to obtain excellent results also by the use of the other processes of combustion. Just how the various advantages and disadvantages will manifest themselves in actual practice is hard to say off-hand. It should be distinctly mentioned, however, that the excellent showing of the Diesel engine is not due to the attempted close approximation to the Carnot cycle, but (since pressure rather than temperature is the determining factor) that its practical success is a result of the introduction and use of high pressures."

In Table 5, V stands for the cylinder volumes required by 1 kg. of gas, the meaning of the rest of the notation being the same as before. As all figures are relative only, this table has not been transposed to English units. A study of the figures given again shows that the methods of operation marked (*a*) and (*b*) are of nearly equal value, while method (*c*), that is, combustion at constant temperature, as laid down in Diesel's "Rational Heat Engine," and covered by his patent, gives thermal efficiencies less by from 15–20% than the results obtained by the other two cycles under the same conditions. It is therefore not surprising to note that the later Diesel machines, operating with combustion at constant pressure, show a greater efficiency than the early machines attempting to use so-called isothermal combustion.

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[1] Schöttler, Die Gasmaschine, 3rd ed., p. 228.

TABLE 5

COMPARATIVE VALUES OF η_t AND V FOR VARIOUS METHODS OF OPERATION

		Maximum Pressure 50 atm.				Maximum Pressure 100 atm.			
		Kgs. of Air to 1 Kg. of Gas.		Kgs. of Air and Burned Gas to 1 Kg. of Gas.		Kgs. of Air to 1 Kg. of Gas.		Kgs. of Air and Burned Gas to 1 Kg. of Gas.	
		50	100	50	100	50	100	50	100
(*a*) Constant-volume cycle. .	$\eta_t=$	0.64	0.66	0.59	0.63	0.72	0.72	0.68	0.71
	$V=$	21+39	41+57	47	89	21+35	41+54	45	86
(*b*) Constant-pressure cycle .	$\rho=$	1.74	1.38	1.75	1.38	1.61	1.31	1.61	1.13
	$\eta_t=$	0.68	0.68	0.63	0.63	0.74	0.74	0.71	0.72
	$V=$	21+37	41+57	45	87	21+34	41+54	44	85
(*c*) Constant-temp. cycle. . . .	$\eta_t=$	0.52	0.61	0.43	0.58	0.64	0.69	0.59	0.67
	$V=$	21+44	41+60	45	87	21+39	41+56	44	85

2. Prof. Donat Bánki, in a comprehensive article "Zur Theory der Wärmemotoren,"[1] comes to the following final conclusion:

"From all investigations so far carried out it is without doubt necessary to raise the pressure in the cylinder of all types of engines as high as practicable. We therefore obtain a correct basis of comparison for the different kinds of engines, if we draw our curves with reference to the *maximum pressures* existing in the cycle, that is, existing at the end of the heat supply."

A study of the curves of total efficiency (i.e., economic efficiency η_w) brings out the following facts, important in their bearing upon the design of heat engines:

The total efficiency increases in all cases with the amount of heat furnished per cycle. The curves for η_w, Fig. 7, at first show a rapid rise as the pressures increase,[2] but beyond a certain pressure the change becomes more gradual. For a small quantity of heat supplied (180 B.T.U. per lb. of air) the curve shows a slight decrease; for a heat supply greater than that, however, η_w is steadily increasing in the pressure range investigated.

The curves for the constant-volume engines rise more rapidly than those for the constant-pressure engines; the former therefore intersect the latter. The point of intersection, however, is found further into the field, the greater the heat supply. For a supply of 180 B.T.U. per lb. of air, for instance, it is located near 10 atm. (140 lbs.), for a supply of 360 B.T.U. it is found near 25 atm. (360 lbs.), for 540 B.T.U. near 60 atm. (1000 lbs.), and for 720 B.T.U. the point of intersection is above 100 atm. (1400 lbs.). With due reference to practical pressure limits, 450–600 lbs., we conclude, therefore, that for small supplies of heat the constant-volume engine, and for larger supplies the constant-pressure engine, will show the greater efficiency, always presupposing adiabatic compression.

The curves for isothermal combustion show far lower efficiencies than those for the other two engines using adiabatic compression. Besides this, this cycle has the disadvantage of being adapted to the utilization of comparatively small quantities of heat only. Hence a machine using this method of operation has in any case little to recommend it in practice, and for that reason isothermal combustion will not be further considered.

In engines using isothermal compression, we obtain acceptable efficiency figures only if, in the case of the constant-volume engines, the heat supply is at least 360 B.T.U. per lb. of air, and in the case of constant-pressure engines at least 540 B.T.U. At a heat supply of 540 B.T.U. the efficiency curves for constant-volume engines with adiabatic compression and for those with isothermal compression approach each other very closely; for a heat supply of 720 B.T.U. they nearly coincide. The efficiency curve for the constant-pressure engine with isothermal compression shows inferior results even for a heat supply of 720 B.T.U., and it is further evident from this curve that it is useless to employ compression pressures in excess of 30 atm. (420 lbs.), since the losses not considered in this investigation (leaky piston, valves, etc.) would probably overbalance the slight theoretical gain shown by the curve.

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[1] Z. d. V. D. I., 1898, p. 893.

[2] The position of these curves of efficiency furnishes the explanation for the fact that the earlier constant-pressure engines (Brayton, Simon, etc.), all of which operated with low pressures, show an efficiency so much inferior to the Diesel engine, in which pressures up to 600 lbs. are used. It also makes clear why hot-air engines, in which, besides low-working pressures, we also have small quantities of heat supplied per unit of air, have such insignificant performances to their credit.

Assuming that rich mixtures are used, *we may say that, concerning the effect upon the efficiency of constant-volume engines, it is immaterial whether the charge is cooled during compression or not, as long as, through proper choice of compression space, the explosion pressures in all cases reach the allowable maximum.* Without affecting the efficiency, therefore, it is possible to cool to such an extent as to make the compression curve an isothermal. But in such case it is necessary to make the compression space smaller than for adiabatic compression, in order to obtain the same explosion pressure.

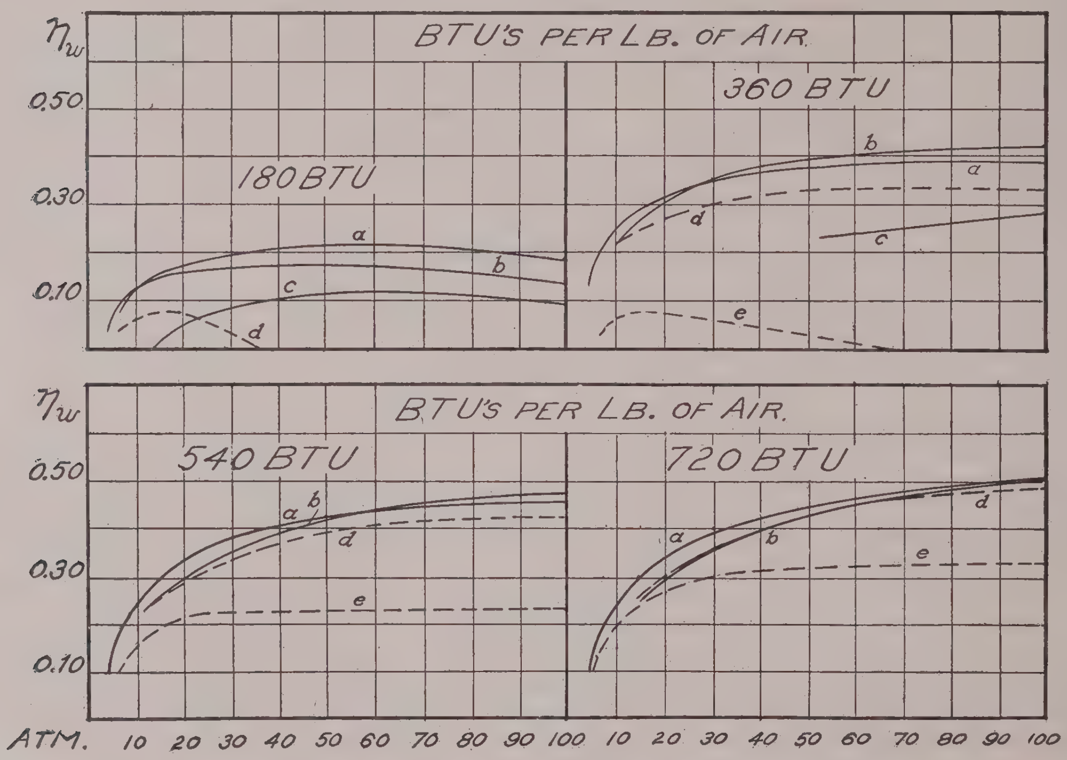

FIG. 7.—Influence of the Heat Content of the Mixture upon Economic Efficiency η_w.

a = Explosion engine with adiabatic compression. b = Constant pressure engine with adiabatic compression. c = Engine with isothermal combustion. d = Explosion engine with isothermal compression. e = Constant pressure engine with isothermal compression.

For constant-pressure engines, the case is quite different. In this type of machine, *cooling during compression has an unfavorable effect upon engine efficiency.*

Now, in practice adiabatic compression is not possible on account of necessary cooling of the cylinder walls. Hence the efficiency curve of the constant-pressure engine with adiabatic compression will approach the curve for the same engine with isothermal compression, that is, the former will in practice nearly coincide with the curves for the constant-volume engine, or may even fall below the latter.

The results of our investigations may be concisely stated under the following heads:

a. The supply of heat should be as large as possible (i.e., use rich mixtures).

b. The method of introducing the heat to the cycle does not affect the efficiency, provided the combustion is somewhere between that at constant volume and that at constant pressure.

c. In constant-volume engines, the charge may be cooled at will during compression; in the constant-pressure cycle, on the other hand, the degree of cooling should be confined to the absolute minimum required for the safety of the cylinder.

d. The maximum pressures in the cylinder should approach 450–600 lbs.

3. Prof. E. Meyer, in a very thorough dissertation entitled "Die Beurteilung der Kreisprocesse," does not, like the two investigators previously quoted, start with equal *maximum* pressures in the cycle, but proceeds on the assumption that the *compression* pressures are the same in all cases. He considers each cycle (after the manner of Zeuner and others) as the sum of a number of infinitesimal, perfect, elementary cycles, each formed by two adiabatic and two isothermal lines; and on this basis develops a treatment of great simplicity and clearness which applies with equal uniformity to all types of cycles.[1]

From a study of the cycle of Carnot, together with the fundamental facts established by the mechanical theory of heat (first and second law of thermodynamics), it is easy to see that where work is performed by a quantity of heat falling from a higher to a lower temperature level, the amount of work so developed depends only upon the quantity of heat concerned and upon the difference in temperature between the two levels. For the purpose of a critical comparison of the various cycles, however, only that Carnot cycle is suited for which the quantity of heat dQ supplied along the upper isothermal (temperature T_1) and the quantity of heat dQ_2 discharged along the lower isothermal (temperature T_2) are infinitely small, so that the two adiabatics are infinitely close together. Under such conditions, it is possible to divide any cycle into an infinite number of elementary cycles, by means of a series of adiabatics, Fig. 8. With insignificant error, we may assume that the various heat elements dQ_1', dQ_1'' are each supplied at constant temperature, and that the heat elements dQ_2', dQ_2'' are each discharged at constant temperature. Then the equations applying to the Carnot cycle in general of course apply to the elementary cycles, and we may write for say the nth cycle, using Zeuner's notation, that

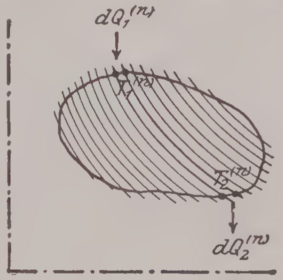

FIG. 8.

$$AdL = dQ_1{}^{(n)} \frac{T_1{}^{(n)} - T_2{}^{(n)}}{T_1{}^{(n)}}, \quad \text{and} \quad \eta_{(n)} = \frac{T_1{}^{(n)} - T_2{}^{(n)}}{T_1{}^{(n)}}.$$

From this we at once derive the following important rule for the condition of best efficiency in a heat engine. Each heat element supplied to a working fluid operating in a cycle should be supplied to the cycle at the highest possible temperature, and each heat element discharged from the cycle should be rejected at the lowest possible temperature. In other words, the endeavor should be to make the temperature limits as wide as possible in each of the elementary cycles. This is the fundamental requirement that must be fulfilled. But it should be noted that, in general, this requirement does not mean that the total heat

$$Q_1 = dQ_1' + dQ_1'' + dQ_1''' + \ldots$$

supplied the cycle should be supplied at a single temperature, i.e., along an isothermal with finite expansion, and that the total heat

$$Q_2 = dQ_2' + dQ_2'' + dQ_2''' + \ldots$$

discharged from the cycle should be rejected at a single temperature, i.e., isothermally. This would result in a Carnot cycle with finite expansion. Almost without exception the various heat elements composing the total heat supplied a cycle are not at our command at a single constant temperature, and in the endeavor to widen the temperature limits of each individual elementary cycle as far as possible, this very fact would lead one to constantly change the temperatures of supply and of discharge. Only in the case that all of the heat units supplied to and all of those discharged from a cycle are available at constant temperature, i.e., when the "sources of heat" are at constant temperature, a case which in the theory of heat engines is the exception, may the Carnot cycle be said to be the ideal.

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[1] Z. d. V. D. I., 1897, p. 1108.

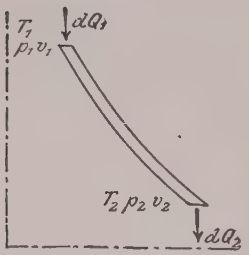

FIG. 9.

We will next consider the question how in a gas engine we may obtain the greatest possible efficiency, and what method of combustion is best adapted to this end. In this discussion it is assumed that the gas-engine cycle is a closed cycle, and that the heat Q_1 is itself introduced into the cycle from an external source, but it is hardly necessary here to prove why any of the assumptions are permissible without great error. Under these conditions the best criterion for the value of the various cycles is again obtained by dividing each into a number of elementary Carnot cycles. Let Fig. 9 represent one of these cycles, composed of two adiabatics and two infinitesimal isothermals. With the notation of Fig. 9, we may write for a perfect gas that

$$T_1 = \frac{p_1 v_1}{R}.$$

Now during the introduction of the infinitesimal heat element dQ_1, the values of p_1 and v_1 undergo but infinitesimal changes, hence we may consider these values the same at the end of the isothermal change without great error. The same reasoning applies to the expression $T_2 = \frac{p_2 v_2}{R}$.

Therefore the efficiency of the elementary cycle under discussion is

$$\eta = \frac{T_1 - T_2}{T_1} = \frac{p_1 v_1 - p_2 v_2}{p_1 v_1} = 1 - \frac{p_2 v_2}{p_1 v_1},$$

and finally, since from the adiabatic law $p_1 v_1^k = p_2 v_2^k$, we obtain [1]

$$\eta = 1 - \left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}} = \frac{p_1^{\frac{k-1}{k}} - p_2^{\frac{k-1}{k}}}{p_1^{\frac{k-1}{k}}}.$$

The above expression, however, may also be considered as having been derived by the use of an elementary cycle in which the heat is supplied at a constant pressure p_1 and rejected at a constant pressure p_2. Depending upon which of the two methods of division gives the clearer insight into the case, the choice may therefore be had between this type of elementary cycle or that of Carnot.

To prevent any misconception, however, it should be noted at this point that the expression $\eta = \frac{T_1 - T_2}{T_1}$ is derived by aid of the fundamental laws of thermodynamics and the fact that temperature is a measure of heat energy, and is consequently applicable to all substances without exception; while the equation $\eta = \frac{p_1^{\frac{k-1}{k}} - p_2^{\frac{k-1}{k}}}{p_1^{\frac{k-1}{k}}}$ is derived from the former by use of the formula $pv = RT$ and the adiabatic law, and is therefore applicable to gases only. Note especially that for the same pressure limits p_1 and p_2, the efficiency η depends upon the nature of the gas, which fact is expressed in the value of the exponent $k=\frac{c_p}{c_v}$. The lowest value which may be assigned to the temperature T_2 is that of the atmosphere; even if the working fluid should reach this temperature under a pressure higher than atmospheric, so that if by further expansion a lower temperature might easily be reached in the cylinder, this would not improve the efficiency. On the other hand, it is of advantage to have the pressure p_2 as low as possible, even much lower than atmospheric pressure, as long as the corresponding temperature is higher than that of the surrounding air. This is another point that should be carefully kept in mind when using the two kinds of elementary cycles.

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[1] The exponent k here used by Meyer is the same as our $x = \frac{c_p}{c_v}$.

From the foregoing discussion it follows that, in order to obtain the greatest efficiency from a cycle, it is necessary to supply each heat element at a pressure p_1, and to reject the part of each element not utilized at a pressure p_2, so that the ratio $\frac{p_1}{p_2}$ shall be as large as possible. From this again we may conclude that, given two fixed pressure limits, that cycle is the most efficient in which the heat is all supplied at constant pressure at the upper limit and the heat not transformed is all discharged at constant pressure at the lower limit, just as the Carnot cycle is best for fixed temperature limits. It is therefore quite illogical to assume that either one of these cycles can be the ideal in all instances, and to use either one of them to determine the available energy for all cases.

In the case of gas engines there is one other point that should be specially noted. As already pointed out, the lower limits of pressure in heat engines at which heat is given off to the cooling water, are not in general set by any physical reasons as long as the temperature of the working fluid at these pressures is higher than that of the cooling water. In steam engines, for instance, in which the lower temperature is produced by water injection into the working fluid, the rejection of heat occurs actually under a considerable vacuum. The case of the gas engine (with the exception of the atmospheric gas engine) is quite different. In this machine a lowering of the pressure in the cylinder below that of the atmosphere is always undesirable because the burned gases must be forced out into the air. The production of the lower temperatures in gas engines by means of water injection has not yet been tried, and in any case the method would seem to offer considerable difficulties. In practice, therefore, in the gas engine we have a fixed lower pressure: p_2 can at no time become appreciably smaller than atmospheric pressure. Hence the requirement for best efficiency in the gas engine may be restated as follows: Each heat element introduced into the cycle must be furnished at the highest pressure possible, and the total heat rejected should be discharged if possible at atmospheric pressure. This statement explains the great advantage of compression in the gas engine.

The reason for this advantage may also be made clear in another way. Of the two requirements for best efficiency in a gas engine, i.e., heat supply at the highest and heat rejection at the lowest possible temperature, the former is fulfilled with very great ease. For even at atmospheric pressures we obtain temperatures during combustion as high as 2700° F. and these in themselves are certainly high enough to guarantee a fair degree of efficiency. On the other hand, there is difficulty in rejecting the heat at temperatures that are low enough. At the present state of development of the gas engine, low temperatures of rejection can only be obtained by allowing the working fluid, after it has attained its highest temperatures, to expand adiabatically as far as possible. But since the lower limit to this expansion is set by the pressure of the outside air, nothing remains but to supply the heat to the cycle at the highest possible pressures.

Bearing in mind the knowledge thus gained, it is now easy to answer the question as to which type of pressure line promises the best efficiency. To this end we compare the process of combustion at constant volume, Fig. 10, with that at constant pressure, Fig. 11, and that at constant temperature, Fig. 12, and it is assumed in each case that at the beginning of combustion the mixtures shall have the same temperature T_c and the same pressure p_c. It is further assumed that in each case the expansion is carried to atmospheric pressure. Now divide each one of the cycles by a series of adiabatics into an infinite number of elementary cycles (for the purpose of clearness only three of these are indicated in each case). Each member of the series of elementary cycles receives consecutively the equal heat elements dQ_1', dQ_1'', $dQ_1''' \ldots dQ_1^{(n)}$. An examination of the figures then discloses the following:

In the process of combustion at constant volume, each heat element dQ_1', dQ_1'' . . . , introduced into the cycle, is utilized to raise the temperature and especially the pressure for the introduction of the next succeeding element. Since, however, in all of the elementary cycles, expansion is carried to the same final pressure (atmospheric) this process of combustion tends of itself to widen the pressure limits for each succeeding cycle, and thus each elementary cycle has a higher efficiency than the one preceding it.

In the case of combustion at constant pressure, only a part of each heat element dQ_1', dQ_1'' . . . is utilized to raise the temperature for the next succeeding element. But since the pressure at which all of the elements are introduced remains the same, the efficiency also remains constant from elementary cycle to elementary cycle, each efficiency being the same as that of the first elementary cycle in the cycle using combustion at constant volume.

The case is even less favorable for combustion at constant temperatures. Here during the introduction of each heat element the pressure rapidly decreases, so that the pressure limits become narrower from cycle to cycle. The efficiency of the elementary cycles therefore constantly decreases as combustion proceeds because the ratio of expansion is growing constantly less.

It follows directly from what has been said that, starting with equal compression pressures, combustion at constant volume shows by far the better efficiencies, while combustion at constant temperature shows very inferior results.

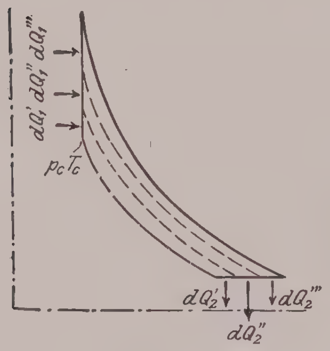

FIG. 10.

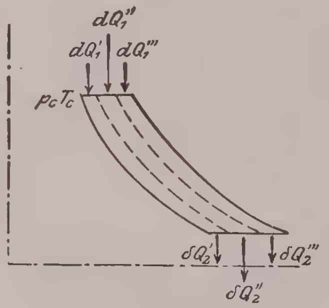

FIG. 11.

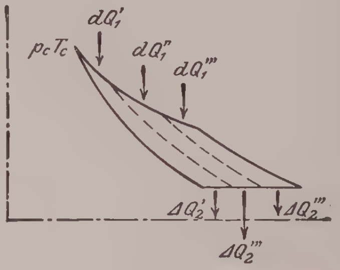

FIG. 12.

It is also easy to see, with the notation of Figs. 10, 11, and 12, and assuming that $dQ_1' = dQ_1'' = dQ_1''' = dQ_1^{(n)}$, that the following equations must hold:

$$dQ_2' = \delta Q_2' = \Delta Q_2',$$

$$dQ_2' > dQ_2'' > dQ_2''' \dots > dQ_2^{(n)} > \dots,$$

$$\delta Q_2' = \delta Q_2'' = \delta Q_2''' = \dots \delta Q_2^{(n)} = \dots,$$

$$\Delta Q_2' < \Delta Q_2'' < \Delta Q_2''' < \dots \Delta Q_2^{(n)} < \dots,$$

and from these it again follows that combustion at constant volume is more efficient than that at constant pressure, and this in turn gives better results than isothermal combustion.

On account of the fact then, as shown above, that, for any given compression pressure p_c, combustion at constant volume gives the best results, our aim should be to make both p_c and T_c as high as possible, so that even the first heat element dQ_1' shall be introduced at high pressure, and then even the first elementary cycle will show a high efficiency. But if the compression is constantly increased, we soon reach limits set by conditions of actual practice. We must therefore next consider both pressure and temperature limits, and such considerations will to a great extent change the views obtained on purely theoretical grounds. Temperature limits, however, play but a subordinate part. Experience has shown that a gas engine may be successfully operated even if the temperatures developed in the cylinder exceed 2900° F. Of course with such temperatures it is necessary to thoroughly cool the cylinder bore, valves, seats, etc., and this introduces another direct loss of heat, which, however, is of an entirely different nature from the quantity of heat necessarily discharged at the lower temperature limit, since by it a certain quantity of heat is even prevented from entering the cycle. But, in spite of this, if temperatures occur along the compression line in excess of the ignition temperature of the gas mixture (appr. 1000° F.), it becomes necessary to cool if it is desired to develop an adequate quantity of heat in the cylinder. Schröter has shown that the heat losses to the cooling water in Diesel engines are about the same as these losses in constant-volume engines (approximately 40% of the heat of combustion), in spite of the fact that in the former the temperatures are kept uniformly lower. But, to repeat what has been said, the upper temperature limits do not affect the general result in any marked degree, because they are, in general, certainly high enough to warrant a much better efficiency than we obtain at the present time if it were only possible to depress the lower temperature limit. And since, to succeed in the latter respect, high initial pressures are necessary, the pressure limits are of the greatest importance. The maximum pressure in the cycle is the factor that determines in the first place whether any given design is constructively possible or not, whether the machine is cheap or costly to build, whether it is easy or difficult of operation, and whether it is sufficiently reliable in operation. The pressures at present occurring in Diesel engines may be considered to give an idea, based on practical experience, of how high we can at present carry maximum working pressures. In this connection there is a further difference between temperature and pressure limits. How high the temperature limit may be put depends largely upon the time during which the maximum temperatures last. The shorter the time, the higher may be the limit for temperature effects are a function of time. On the other hand, all of the machine parts must be built to be safe under the maximum pressure, whether the duration of this pressure is instantaneous only or distributed over a longer period. It may be said to be the fact even that pressures lasting only an instant are more severe than the same pressures acting during a longer interval.

Now suppose that, instead of assuming equal compression pressures in all cycles, we assume equal maximum pressures. Then for combustion at constant volume, the pressure at the end of combustion must be equal to this maximum pressure. Under these conditions, only the last of the heat elements is introduced at the most favorable maximum pressure, while all of the other elements are introduced at lower, i.e., less favorable pressures. For combustion at constant pressure, assuming that the working fluid is adiabatically compressed to the maximum pressure, each and every heat element is introduced at this most efficient pressure. But it must also be evident that combustion at constant temperature gives results very much inferior to either. For at the beginning of the isothermal combustion process we have the maximum pressure, for which the machine parts must be designed, from which we obtain no advantage, since even during combustion the pressure decreases so rapidly that for every succeeding elementary cycle the ratio of expansion is considerably less, resulting in decreased efficiency. The net result, therefore, assuming a given pressure limit, is that, providing it is possible to compress up to this maximum pressure, combustion at constant pressure is the most efficient. Since under these conditions the temperature increases during the combustion period, it would seem to follow theoretically that during this time the loss to the cooling water should be somewhat greater than during isothermal combustion; but according to the experience of Diesel, who with isothermal combustion obtained hardly any development of work diagram, this fact does not have much weight.

It is evidently not correct, in answering the very general question as to the most favorable cycle for heat engines, to designate certain definite cycles of operation as undoubtedly the best or even "ideal," especially as far as the possibility of practical operation is concerned. For the choice in every case should depend upon the special governing conditions. The best method of procedure for the designer would therefore seem so be to subdivide the pressure diagram which he desires to carry out in his proposed machine, into elementary cycles by the method above discussed, and then by an examination to see how each elementary cycle might possibly be further improved by raising p_1 and T_1, or by depressing p_2 and T_2. Thus for instance the rule that combustion at constant pressure is the most efficient does not apply to Otto's 4-cycle machine, because in this case the allowable limit of compression does not coincide with the maximum pressure limit. The latter may easily exceed 20 atm. (300 lbs.), but the former must not exceed 7–8 atm. (100–125 lbs.) if pre-ignition in the gas-air mixtures is to be prevented.

In such a case then the desire to introduce each heat element at the highest pressure possible, naturally leads to the constant-volume cycle, since in it pressures increase most rapidly.[1]

C. CRITICAL EXAMINATION OF THE VARIOUS CYCLIC EVENTS

The preceding pages contain an investigation of the various operating cycles as a whole, according to the laws of thermodynamics, i.e., with reference to the complete operation of the transformation of heat into work. It next becomes necessary to examine into the various events composing each cycle in so far as actions and interactions occurring are of interest to the designer and constructor. In the following discussion, the general notation used in the previous pages has been retained, the various states of the working fluid being indicated by proper indices in the accompanying diagrams.

I. The 4-cycle Events

1. The Suction Stroke, Fig. 13. At the inner piston position, the exhaust stroke has just been completed, and the clearance space is filled with V_c cu.ft. of burned gases under an absolute temperature of T_r° F., and an absolute pressure of P_r in pounds per square foot. The weight of this volume in pounds is

$$G_r=\gamma_r V_c=\frac{V_c P_r}{T_r R_r} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (46)$$

if γ_r represents the specific weight of the gases (in pounds per cubic foot), and R_r is the gas constant.

If now the piston begins its outstroke, the clearance gases will re-expand until the pressure falls from P_r to P_a; only after this pressure is reached, fresh mixture (or in

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[1] From the general formula for the elementary cycle, Fig. 9, $\eta=\frac{T_1-T_2}{T_1}$. We may also derive for gases

$$\eta=1-\left(\frac{v_1}{v_2}\right)^{k-1}=\frac{v_2^{k-1}-v_1^{k-1}}{v_2^{k-1}}.$$

An examination of this equation brings out the following points: In order to utilize the heat supplied to a gas engine to the best advantage every heat element should be introduced at the smallest volume possible, and the part not utilized should be rejected at the largest volume possible. If, therefore, in a gas engine we have fixed volume limits, the most efficient cycle is composed of two adiabatics and two lines of constant volume. In the Otto 4-cycle engine we actually have these fixed volume limits. The maximum volume is equal to the total volume of the charge drawn in at atmospheric pressure and temperature; the minimum volume is determined by the consideration of pre-ignition of the charge. Consequently the best efficiency is attained when the cycle receives and rejects the respective quantities of heat only at the respective volume limits. In the case of the Diesel engine, we have given for the reception of heat a pressure limit, and for the rejection of heat a volume limit (since the expansion can proceed only to the volume at which the charge has atmospheric pressure and temperature). In this engine, therefore, the heat is rejected at constant volume, in which fact it differs in its operation from the constant-pressure cycle above discussed, in which expansion is carried to atmosphere; but it is easy to see that under the conditions imposed combustion at constant pressure will show the best efficiency.

the Diesel only air) enters, mixing with the burned gases and thereby becoming heated. Assuming that the external temperature is T_g, that the barometer pressure is 30″ Hg. (14.7 lbs. per sq.in.) $= P_g$, and that the real volumetric efficiency of the stroke $= \eta_e$, the mixture drawn in will weigh

$$G_g = \gamma \eta_e V_h = \frac{\eta_e V_h}{T_g} \cdot \frac{144 \times 14.7}{R_g} \text{ pounds.} \quad (47)$$

At the end of the suction stroke the weight of the total charge will be

$$G_a = G_r + G_g = \frac{V P_a}{T_a R_a} \text{ pounds} \quad . \quad . \quad . \quad (48)$$

in which T_a is the absolute temperature of the total charge in degrees F., P_a the corresponding pressure in pounds per square foot, and $V = V_c + V_h$.

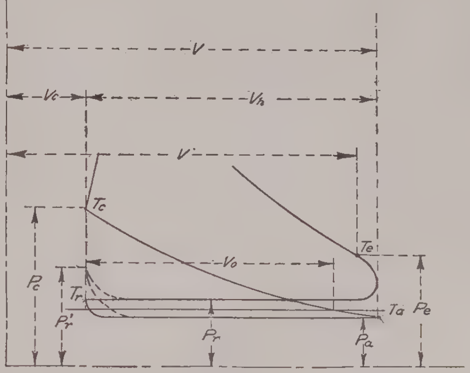


FIG. 13.

If we assume equal values for the three gas constants, R_r, R_g, and R_a (for burned gases, fresh mixture, and for the total charge) which can be done without sensible error, we may write

$$G_a R_a = \frac{V P_a}{T_a} = \frac{V_c P_r}{T_r} + \frac{2117 \eta_e V_h}{T_g}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (48a)$$

and from this the actually drawn-in volume of the mixture, referred to a pressure of 14.7 lbs. per square inch, and a temperature of $T_g°$, is

$$V_h' = \eta_e V_h = \left(\frac{V P_a}{T_a} - \frac{V_c P_r}{T_r} \right) \frac{T_g}{2117} \text{ cu.ft.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (49)$$

Distinction should at this point be made between the real volumetric efficiency η_e and what may be called the apparent volumetric efficiency η_v. Referring to Fig. 13, the latter may be expressed by

$$\eta_v = \frac{V_o}{V_h}.$$

while the former represents the ratio of the volume V_h' of the charge at a pressure P_g and a temperature T_g, actually drawn into the piston displacement V_h, that is

$$\eta_e = \frac{V_h'}{V_h}.$$

The volume V_h' is less than V_o on account of the fact that the temperature in the cylinder is greater than T_g, and hence the real volumetric efficiency η_e is always smaller than the apparent volumetric efficiency η_v obtained by ratio of volumes from a weak spring card without temperature correction.

Assuming an atmospheric temperature of $t = 62°.5$ F., i.e., $T_g = 523°.5$ F., the real volumetric efficiency η_e of the suction stroke will then be theoretically

$$\eta_e = \frac{\left(\frac{V P_a}{T_a} - \frac{V_c P_r}{T_r} \right) \frac{T_g}{2117}}{V - V_c} = \left(\frac{\varepsilon P_a}{T_a} - \frac{P_r}{T_r} \right) \frac{.247}{\varepsilon - 1} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (51)$$

in which $\varepsilon = \frac{V}{V_c} =$ ratio of compression as before.

The value of η_e therefore depends largely upon P_a and P_r, both of which may be easily determined by means of a weak spring. Temperature T_r is a function of the exhaust and jacket water temperatures, while T_a depends mainly upon ε and the jacket temperature. If we assume that the gas constants R_r, R_g, and R_a are equal, it follows from eq. (48a) that

$$T_a=\frac{VP_a}{G_aR_a}=\frac{VP_a}{\dfrac{2117\eta_eV_h}{T_g}+\dfrac{V_cP_r}{T_r}},$$

and again putting $T_g=523°.5$ F. this reduces to

$$T_a=\frac{.247VP_a}{\eta_eV_h+\dfrac{.247V_cP_r}{T_r}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (52)$$

Note that if in any given case the barometer pressure differs considerably from 30″ Hg. (14.7 lbs. per sq.in.), the value of the factor 2117 in the above derivation must be altered correspondingly.

The derivation of eqs. (49)–(52), however, depends upon assumptions and conditions which do not quite hold true or exist in actual practice. Thus, for instance, the metallic walls act as regenerators of heat; they absorb a considerable quantity of heat during combustion and expansion, and transmit a part of this heat immediately to the cooling water, transferring the remainder afterwards to the next incoming charge. Even if we assume that the mean wall temperature is equal to the outlet temperature of the cooling water, there will still be a temperature difference of from 60–90°, between the walls and the incoming charge. In reality, however, the walls are much hotter than the cooling water, because even from the start a complete transfer of all the heat absorbed by them during every combustion and expansion process is not possible. The final temperature of the walls at which equilibrium is maintained between the heat absorbed and that discharged must therefore be higher than the temperature of the cooling water. In small engines with thin cylinder walls this difference may be assumed to be at least from 45–55° F., in large cylinders it may be from 60–110° F. and more (see p. 121, under "Cylinders"). On this account the temperature difference between walls and fresh charge is likely to be from 150–180°, and it is evident that owing to the consequent flow of heat, the charge must be considerably heated up, causing a corresponding increase in T_a. This action explains the well known fact that with increasing load on an engine, the value of η_e decreases.

Concerning the temperature T_r of the exhaust gases remaining in the clearance space, we know nothing definitely. It is certain only that it is not lower, but probably higher, than the temperature of the gases in exhaust, and T_r probably also rises and falls with T_ϵ. Now since T_e is on the average 1540° F. (and over), it follows that the clearance gases must even during the suction stroke transfer a considerable quantity of heat to the cylinder walls which are cooler by from 700–900°. This action causes a condensation of the burned gases, which in itself affects the volumetric efficiency η_e favorably, since the contraction volume practically adds to the stroke volume by the same amount.

The value of η_e is of importance to the designer, especially in the determination of the cylinder dimensions required for any given engine capacity (see p. 72, Part III). But since it is not possible to determine η_e mathematically, we are compelled to fall back on actual measurements of the real volume of mixture V_h', drawn in per stroke.

In the case of small gas engines such measurements are comparatively easily made with the aid of air and gas meters, and they have been occasionally so made in scientific investigations; for large engines the volume measurements usually require apparatus so bulky and costly that the attempt is not often made. Engineering literature consequently offers but little generally applicable information on this point. The table below gives a few average values of η_e, from reliable data furnished by Slaby, Meyer, and others, taken either directly or determined by computation from the figures given.

The remarks made concerning the values of P_r and P_a in eqs. (48)–(51) apply also in practice, that is, both should be kept as close as possible to the atmospheric line. Especially severe are the effects of an increase in P_r, since it increases the weight of the exhaust gases remaining (by compressing them from P_r to P_r'), and this in turn decreases the volumetric efficiency η_e, owing to the re-expansion of the gases from P_r or P_r' to P_a. Through a too early closure of the exhaust valve, P_r' may be made considerably greater than P_r with a consequent serious decrease in engine capacity.

As previously shown, the apparent volumetric efficiency may be found from the equation

$$\eta_v = \frac{V_o}{V_h}.$$

What values of η_v may occasionally be found is well shown by Fig. 14, which represents a weak spring card taken by the author from an automobile engine. Without reference to any temperature differences, this diagram shows a value of η_v=appr. 60%.

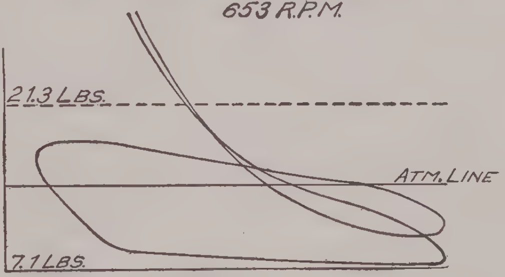


FIG. 14.

Practical Values of p_r, T_r, p_a, and η_e. For the latter the temperature of the charge has been reduced to 32°.

$$p = \frac{P}{144} \text{ pounds per sq.in.}$$

p_a=12.8 to 13.7 lbs./sq.in., η_e=.88 to .93 for slow-speed engines with mechanically operated inlet valve.

p_a=12.5 to 13.3 lbs./sq.in., η_e=.80 to .87 for slow-speed engines with automatic inlet valve.

p_a=11.8 to 12.5 lbs./sq.in., η_e=.78 to .85 for high-speed engines with mechanically operated inlet valve.

p_a=11.5 to 12.2 lbs./sq.in., η_e=.65 to .75 for high-speed engines with automatic inlet valve.

p_a= 8.8 to 11.0 lbs./sq.in., η_e=.50 to .65 for very high-speed auto-engines with automatic inlet valve and air-cooling.

Suction generators and carburetors in unfavorable cases may decrease the above values of η_e by from 2–5%.

$$\left.\begin{array}{l} p_r=15.7 \text{ to } 17.0 \text{ lbs./sq.in.} \\ T_r=1180° \text{ to } 1350° \text{ F.} \end{array}\right\} \text{Too early closure of the exhaust valve and very long or too small exhaust pipe may increase these values considerably.}$$

2. The Compression Stroke, Fig. 13, p. 29. The returning piston compresses the charge up to atmospheric pressure and beyond this to the final compression pressure P_c at the end of the in-stroke. The compression line may be considered as a curve following the general law $PV^n=\text{const.}$ With this assumption, the pressure at the end of compression will then be

$$P_c=P_a\left(\frac{V}{V_c}\right)^n=P_a\varepsilon^n, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (53)$$

while the final temperature is

$$T_c=T_a\left(\frac{P_c}{P_a}\right)^{\frac{n-1}{n}}=\frac{T_aP_c}{P_a\varepsilon}=T_a\varepsilon^{n-1}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (54)$$

For any given P_c or T_c the clearance volume must be

$$V_c=\frac{V}{\varepsilon}=V\left(\frac{P_a}{P_c}\right)^{\frac{1}{n}}=V\left(\frac{T_a}{T_c}\right)^{\frac{1}{n-1}}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (55)$$

From the well known equation $PV=RTG$, we may also write

$$P_c=\frac{T_cG_aR_a}{V_c} \quad (53a) \qquad T_c=\frac{P_cV_c}{G_aR_a} \quad (54a) \qquad \text{and} \qquad V_c=\frac{T_cG_aR_a}{P_c}. \quad (55a)$$

TABLE 6

ABSOLUTE COMPRESSION PRESSURES (p_c) AND TEMPERATURES (T_c) FOR DIFFERENT VALUES OF ε, n, AND T_a.

Ratio of Compression $\varepsilon=$			2.0	2.5	3.0	3.5	4.0	4.5	5.0	6.0	7.0	8.0	9.0	10.0
$n=1.25$	For $p_a=12.5$,	$p_c=$	29.7	39.2	49.3	59.9	70.7	81.8	93.2	117.5	142.0	168.0	195.0	222.0
	when $T_a=$ 600	$T_c=$	690	720	747	770	792	811	828	858	885	910	930	951
	650		748	780	811	835	858	877	896	930	958	983	1010	1028
	700		805	840	875	900	924	945	965	1002	1032	1062	1088	1110
	750		863	900	937	956	990	1013	1036	1073	1107	1137	1163	1188
	800		921	960	1000	1027	1056	1080	1105	1144	1180	1212	1240	1267
$n=1.30$	For $p_a=12.5$,	$p_c=$	30.7	41.2	52.0	63.7	75.7	88.3	101.2	128.2	156.7	186.5	217.5	250.0
	when $T_a=$ 600	$T_c=$	703	740	772	800	825	847	868	905	938	966	1000	1018
	650		763	803	837	866	895	917	942	980	1015	1043	1076	1102
	700		821	864	900	933	964	987	1012	1055	1093	1128	1160	1187
	750		880	925	965	1000	1030	1058	1087	1130	1170	1210	1242	1272
	800		937	987	1028	1065	1100	1130	1157	1206	1250	1290	1322	1355
$n=1.35$	For $p_a=12.5$,	$p_c=$	31.9	43.0	55.1	67.8	81.2	95.2	109.5	140.5	162.4	206.5	242.2	300
	when $T_a=$ 600	$T_c=$	718	760	797	832	860	885	912	955	995	1030	1063	1092
	650		778	825	864	900	932	963	987	1035	1078	1115	1152	1182
	700		838	888	932	970	1002	1035	1064	1114	1157	1200	1240	1275
	750		897	952	997	1038	1075	1105	1139	1193	1242	1284	1328	1364
	800		958	1012	1062	1108	1146	1180	1214	1264	1323	1372	1418	1455
$n=1.41$	For $p_a=12.5$,	$p_c=$	33.1	45.5	58.8	73.1	88.3	103.0	120.6	156.5	194.0	234.0	277.0	321.0
	when $T_a=$ 600	$T_c=$	735	783	826	864	897	927	956	1013	1055	1095	1135	1170
	650		795	847	893	936	972	1003	1033	1097	1140	1187	1228	1268
	700		857	912	962	1008	1048	1080	1113	1180	1228	1278	1320	1363
	750		917	976	1030	1077	1122	1157	1194	1265	1316	1368	1415	1462
	800		977	1142	1098	1149	1195	1232	1273	1348	1402	1458	1510	1558

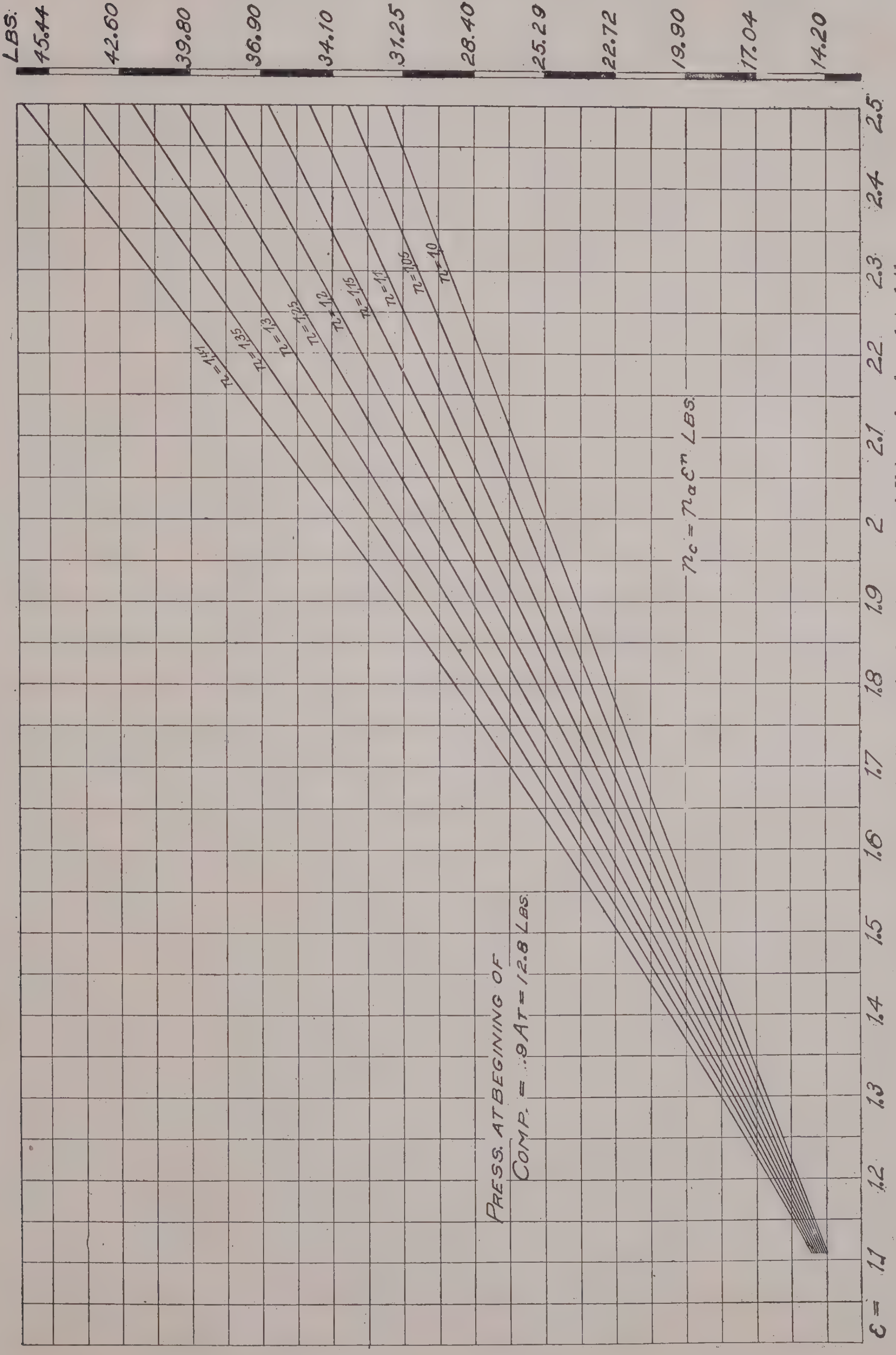


FIG. 15.—Compression Pressures p_c for Values of ε from 1 to 2.5, and Values of n from 1 to 1.41.
(For Pressures and Temperatures at higher degrees of compression, see Plate I.)

Table 6, p. 32, diagram Fig. 15, and Plate I, have been constructed on the assumption that the vacuum at the end of the suction stroke is .1 atm. (1.42 lbs.), that is, that $P_a = .9$ atm. (12.8 lbs.). If in any actual case the real suction pressure should be some other value, as P_a', the readings of the above table and diagrams can be corrected in the ratio $\frac{P_a'}{P_a}$.

Concerning the heat interchanges during the compression stroke, we may note the following: There is little doubt that the charge temperature T_a at the end of the suction stroke differs but little from the mean temperature of the cylinder walls. During the first part of the compression stroke, as long as the outer, cool end of the cylinder is still in contact with the charge, it is likely that the heat of compression is completely taken up by the cooling water, keeping the charge temperature about constant. But the further the charge is compressed into the inner hot end of the cylinder, the faster will the charge temperature increase, and the smaller will be the amount of heat taken up by the cooling water. Even if, at the beginning of the stroke, there should be a heat transfer from the walls to the charge, such movement can be of but short duration and the heat transferred is of small amount. Under ordinary cooling conditions, on the other hand, the heat transfer to the jacket is always considerable, and this fact manifests itself in the indicator cards in that, with increasing compression, the exponent n grows continually smaller, its mean value being considerably under the ratio x of the specific heats.

Values of n from Practice. Owing to the varying heat interchanges along the compression line, as pointed out above, the exponent n is not a constant for the entire line. Its mean value varies from 1.30 to 1.38 in ordinary cases, with an average of about 1.35. But imperfect cooling or high wall temperatures may raise the value of n above the adiabatic value x. Charge losses through leaky pistons and valves give apparent values of n which are too small.

3. The Combustion and Expansion Strokes. Diagram Fig. 16. Ignition should, at least in theory, begin early enough so that the entire charge is burned at the inner piston position. Depending upon the purity, heating value, temperature, and compression of the charge, as well as upon location and nature of the spark, form of combustion chamber, and other practical conditions, the time of ignition and combustion is of varying duration, as indicated by the more or less inclined position of the combustion line or by varying expansion lines.

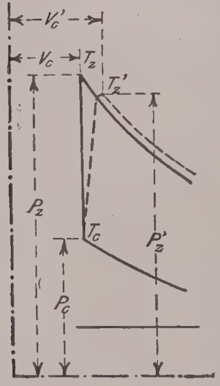
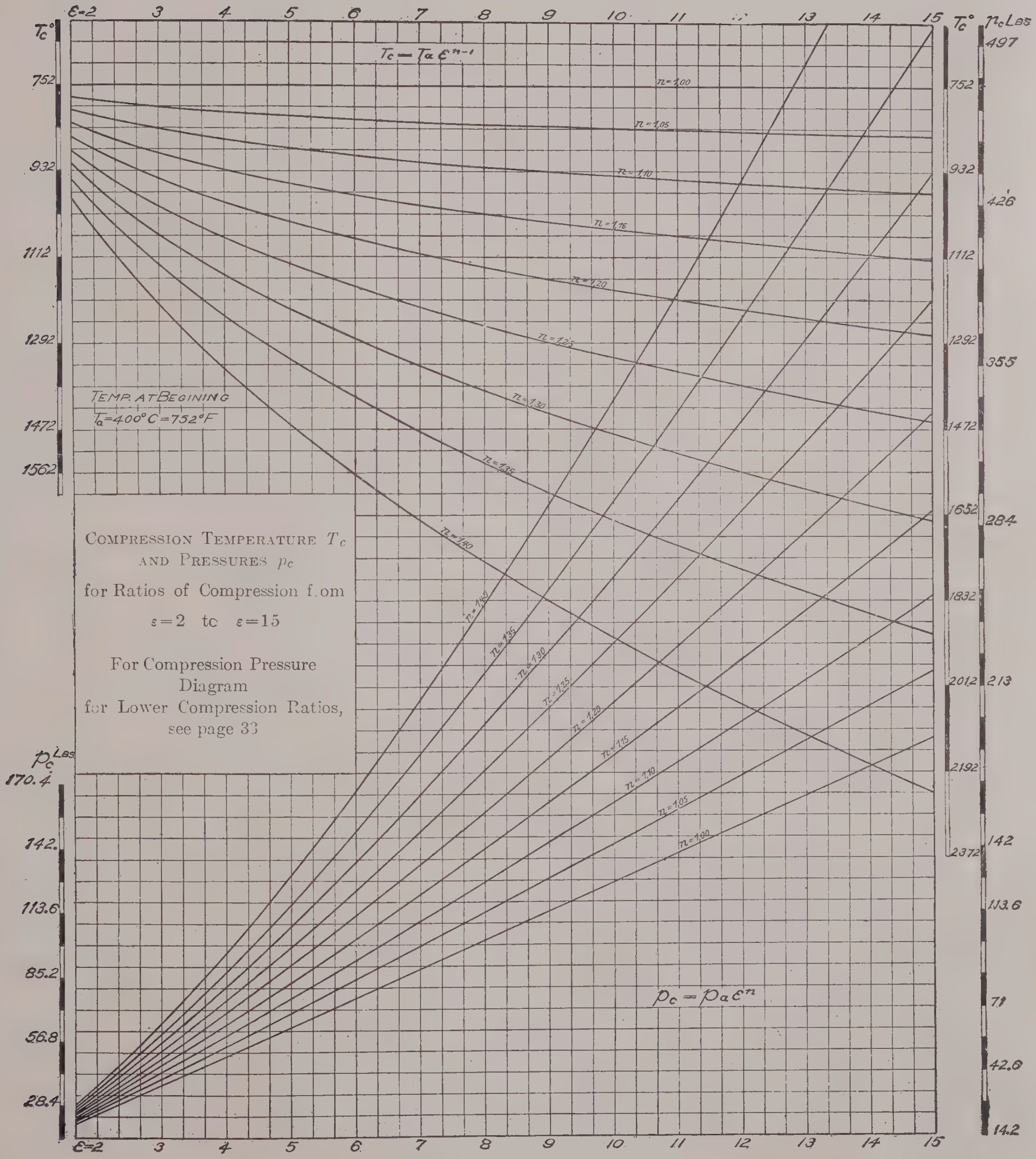


Fig. 16.

The effect of a variation of the point of ignition upon capacity and efficiency may be very different in different machines. It is, for instance, now and then found that a given machine gives the best economic result when the point of ignition is so early as to make the beginning of the explosion clearly distinguishable along the compression line, and every explosion is accompanied by a distinct knock. Most engines, however, cannot stand a spark so greatly advanced, and operate much more smoothly with an ignition point moderately late. It is clear that the diagram area, and with it the engine capacity, may be increased up to a certain point by late ignition, but the reasons for the fact, also sometimes met with, that late ignition favors better utilization of heat, are not so patent.

No definite practical rule for locating the point of ignition can be given. The indicator gives reliable information only when the drum receives a continuous motion,

PLATE I



COMPRESSION TEMPERATURE T_c AND PRESSURES p_c

for Ratios of Compression from $\varepsilon=2$ to $\varepsilon=15$

For Compression Pressure Diagram for Lower Compression Ratios, see page 33 or when a "displaced" diagram is taken, because in the ordinary indicator card, in which the combustion line is placed at the end of the drum motion, the various phenomena are not sufficiently distinct. The best means of locating the most favorable point of ignition is probably to make the ignition gear adjustable, and then to locate the best point by trial.

E. Meyer, in his "Untersuchungen am Gasmotor," has the following to say regarding this question:[1] "The earlier the point of ignition, the greater the heat loss to the walls. The thermal efficiency reaches its maximum with a mean point of ignition, but this point does not need to be very strictly maintained because the consumption of heat differs but little within the limits of 15° of crank angle, and only shows a decided increase for an early point of ignition (20° crank angle below the dead center). As for the rest, late ignition causes a loss of work through spreading of the indicator diagram, but at the same time the loss of heat to the walls is less."

It should be remembered that the above remarks apply to a single definite engine.

Assuming that the combustion is instantaneous, that is, occurs at constant volume, a condition which hardly ever obtains in practice, the maximum explosion pressure will be

$$P_z=\frac{P_cT_z}{T_c}=\frac{T_zG_aR_r}{V_c}. \qquad (56)$$

The explosion temperature is

$$T_z=\frac{P_zT_c}{P_c}=\frac{P_zV_c}{G_aR_r}, \qquad (57)$$

or,

$$T_z=T_c+\frac{Q_1}{c_vG_a}, \qquad (58)$$

when G_a pounds of charge contain Q_1 heat units. Eq. (58), however, cannot be depended upon since the results may be affected by incomplete combustion and other causes.

When the combustion becomes slow through late ignition or other causes, so that $V'_c > V_c$ the above equations change to

$$P'_z=P_c\frac{T'_zV_c}{T_cV'_c}=\frac{T'_zG_aR_r}{V'_c}. \qquad (56a)$$

and

$$T'_z=T_c\frac{P'_zV'_c}{P_cV_c}=\frac{P'_zV'_c}{G_aR_r}. \qquad (57a)$$

If the values of P_z or P'_z can be taken directly from an indicator diagram, eqs. (57) or (57*a*) will give the real combustion temperatures T_z or T'_z. If, on the other hand, P_z is taken from a hypothetical indicator card, the real value of T_z, or, in the reverse case, of P_z can only be found by multiplying by a reduction factor whose value expresses the decrease of temperature or pressure owing to heat losses, cooling, etc., and which value is not far from that of the card factor η_g of the cycle (see p. 7).

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[1] Z. d. V. D. Ing., 1902, p. 1307.

At the moment the exhaust valve opens, i.e., at the beginning of exhaust, the burned gases have expanded to the terminal pressure (see diagram, Fig. 13).

$$P_e = P_z\left(\frac{V_c}{V'}\right)^n = \frac{P_z}{\delta^n}, \quad \ldots \ldots \ldots \ldots \quad (59)$$

in which $\delta = \frac{V'}{V_c}$ represents the real ratio of expansion. At the same time the end temperature is

$$T_e = T_z\left(\frac{V_c}{V'}\right)^{n-1} = \frac{T_z}{\delta^{n-1}}; \quad \ldots \ldots \ldots \ldots \quad (60)$$

or, using G_a and R_r, we also have

$$P_e = \frac{T_e G_a R_r}{V'}, \quad \ldots \ldots \ldots \ldots \quad (59a)$$

and

$$T_e = \frac{P_e V'}{G_a R_r}. \quad \ldots \ldots \ldots \ldots \quad (60a)$$

It would appear from equations (59) and (60) that the greater the ratio of expansion, the lower will be the terminal pressure and temperature, which would make the principle of "complete expansion" appear of great benefit. In 4-cycle Otto engines, however, the ratio of expansion is in most cases nearly equal to the ratio of compression, that is, $\delta \sim \varepsilon$, from which we may write eq. (59),

$$P_e = P_z\left(\frac{V_c}{V}\right)^n = \frac{P_z}{\varepsilon^n}. \quad \ldots \quad (61)$$

This equation shows that an increase in compression also causes a decrease of terminal pressure and temperature, which explains the fact that in our present day of high-compression engines, complete expansion has lost all practical importance.

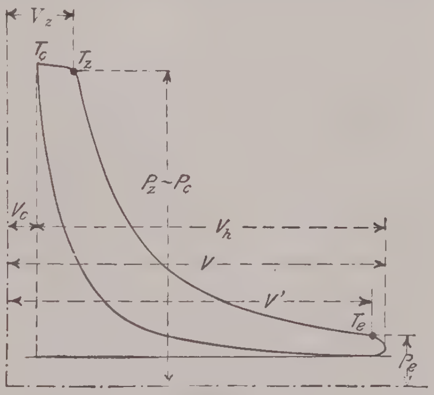

FIG. 17.

In the *constant-pressure cycle*, Fig. 17, the compression pressure P_c is about equal to the maximum combustion pressure P_z. With a cut-off ratio $\rho = \frac{V_z}{V_c}$, and a ratio of expansion $\delta = \frac{V'}{V_z}$ we then have

$$T_z = T_c\frac{V_z}{V_c} = T_c\rho \sim \frac{P_z V_z}{G_a R_r}, \quad \ldots \ldots \ldots \ldots \quad (62)$$

or, assuming complete combustion as before,

$$T_z = T_c + \frac{Q_1}{c_p G_a}. \quad \ldots \ldots \ldots \ldots \quad (62a)$$

For the end-point of the expansion we may write, as in the case of the constant-volume cycle,

$$P_e = P_z\left(\frac{V_z}{V'}\right)^n = \frac{P_z}{\delta^n}, \quad \ldots \ldots \ldots \ldots \quad (63)$$

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and
$$T_e=T_z\left(\frac{V_z}{V'}\right)^{n-1}=\frac{T_z}{\delta^{n-1}}, \qquad (64)$$

or,
$$P_e=\frac{T_eG_aR_r}{V'}, \qquad (63a)$$

and
$$T_e=\frac{P_eV'}{G_aR_r}. \qquad (64a)$$

In both the constant-volume and the constant-pressure cycles, the real *exhaust* temperature is less than the terminal temperature T_e, above computed, because the former is measured outside of the cylinder after complete expansion of the gases to atmosphere. In both cycles the expansion of the burned gases in the cylinder takes place under continued loss of heat to the cooling water, hence the exponents n of the expansion line are considerably greater than the adiabatic ratio x of the specific heats of the charge. At the beginning of the expansion, when the cooling surface of the cylinder is still small and the losses to the cooling water are more or less balanced through after-burning of the charge, the differences between n and x are in most cases quite small. It may even happen that the heat supply due to after-burning balances or overbalances the entire heat loss, in which case n approaches the value of the isothermal exponent. But the greater the cooling surface uncovered by the outgoing piston, the greater the heat losses and the greater the corresponding value of n, although on account of the accompanying rapid fall in the temperature and the consequent smaller loss of heat to the jacket, the rise in the value of n is less rapid near the outer end of the stroke. It also often happens that the value of n varies quite irregularly, alternately increasing and decreasing, so that it becomes impossible to recognize any definite law of variation. An increase in the piston speed is, for obvious reasons, nearly always accompanied by a decrease in the mean value of n for the entire stroke.

For the method of determining n from given indicator diagrams, see Appendix.

Values from Practice. The value of the exponent for the expansion line is generally found between 1.35 and 1.50, although diagrams showing up to $n=1.70$ are not rare. Loss of charge through leakage increases the real value of the exponent; indicators having considerable friction loss have an apparent effect in the opposite direction.

The explosion temperature T_z in our present day high compression engines no longer remains much below 3600° F. abs. (see the entropy diagram, Fig. 2, p. 10). This may be quickly shown by an approximate computation: For $\varepsilon=6$, $n=1.30$, $p_a=12.8$ lbs. per sq.in., and $T_a=684°$ F. abs., we get from Table 6, $p_c=9.24$ atm. (131.5 lbs.) and $T_c=1170°$ F. abs. If we assume that $\frac{p_z}{p_c}\sim 3$, a figure easy to reach for maximum load, we have $p_z=395$ lbs. and from eq. (57) then $T_z=1170\cdot\frac{395}{131.5}=\sim 3500°$ F. abs. The maximum pressure in the cycle is usually also accompanied by the maximum temperature, i.e., the heat supplied by after-burning is less than that used by the expansion and cooling of the gases. The opposite condition is at least no longer common in our modern engines. Assuming a mean exponent $n=1.5$ for the expansion line, the temperature at the end of the expansion in the case chosen would be $T_e=\frac{3500}{6^{0.5}}=\sim 1440°$ F. abs.; for $\delta=3$, however, T_e would have been equal to $\frac{3500}{3^{0.5}}=$approx. 1980° F.

In the case of the constant-pressure engine, if, in agreement with actual conditions, we assume $p_c = p_z = 33$ atm. (470 lbs.), $\varepsilon = 16$, $\delta = 5.5$, $\rho = 3.0$, and further take the exponent n as above, the maximum combustion temperature will be, from eq. (62), $T_z = 1580 \times 3 = 4740°$ F. and the end temperature of expansion, from eq. (64), $T_e = \frac{4740}{5.5^{0.5}} =$ 2040° F. abs. The maximum and mean temperatures of the constant-volume cycle are therefore considerably smaller than those of Diesel's constant-pressure engine.

4. The Exhaust Stroke, Diagram Fig. 13, p. 29. The exhaust valve usually commences to open at about 90% of the expansion stroke. The burned gases escape and the pressure sinks rapidly from P_e to P_r. During the establishment of pressure equilibrium (exhaust) the velocity of the outflowing gases is very high, in the neighborhood of 2600 to 3000 ft. per second,[1] and is independent of the port area. From the kinetic energy of the exhaust gases

$$Q_a = \frac{G_a C^2}{778 \times 32.2 \times 2}, \qquad (65)$$

in which G_a is the weight of the gases in pounds, we may derive the mean exhaust velocity

$$C = 224 \sqrt{\frac{Q_a}{G_a}} \text{ ft. per sec.} \qquad (66)$$

The simplest way of determining Q_a is to consider it as the third member (remainder) in the heat balance, the other two being the heat equivalent of the work and the jacket loss respectively.

The actual exhaust should be completed by the time the piston has reached the dead center, so that on the back-travel it is called upon to overcome only the resistance P_r due to friction in valves and ports. If on the other hand the exhaust is retarded (see Fig. 18, from a high-speed engine), the effect is not only to increase the negative work but also to increase cylinder temperature, which in turn seriously decreases the volumetric efficiency and the allowable compression pressure. Regarding the harmful effect of P_r, see p. 31.

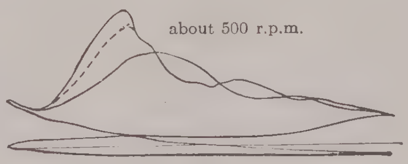

FIG. 18.

If the lower loop of the diagram is taken with a weak spring, remarkable pressure variations are sometimes disclosed. In many cases their cause is not to be sought as much in the engine as in the exhaust line. The exhaust gases, discharged very suddenly and under very high velocity, acquire under certain conditions certain vibratory movements accompanied by corresponding pressure variations. With an exhaust line of sufficient length and no sudden turns, the kinetic energy of the exhaust gas column, initially acquired, may be sufficient at times to overcome all frictional resistances in the line, so that P_r drops to atmospheric pressure, or, owing to overexpanding of the gas, it may even drop below it. Crossley Brothers tried to make this accidental phenomenon one of regular occurrence and attempted to utilize it as a means for scavenging the cylinder. Jul. Söhnlein even went a step further, intending to cause the vibrating gas column to draw in the new charge, in 2-cycle engines.[2]

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[1] See Slaby, Kalorimetrische Untersuchungen, p. 191.

[2] German Patent, No. 83210, 1894.

Neither one of these schemes had any practical success, because the occurrence of the vacuum could not be controlled with certainty, the exhaust action being subject to the influences of too many accidental conditions. With ports sufficiently large and with correctly timed valve opening, the occurrence of a vacuum at the beginning of the exhaust stroke is common, quite probably because at that time the kinetic energy of the gas column is greatest (see diagram Fig. 19, from one of the older types of Crossley engine); in some other cases this vacuum persists until near the middle of the stroke (Fig. 20, from a Diesel engine); or it may not occur at all (Fig. 21, card from a Priestman engine). If the frictional resistances encountered in the exhaust pipe are of such a nature as to set the gases into vibration, the vacuum may occur intermittently, the line cutting the atmospheric pressure line repeatedly (Fig. 22, diagram from an Otto engine). It was shown by E. Meyer in the Zeitschrift d. V. D. I., 1901, p. 1343, that this wave form of the exhaust line is not always due to inertia effects in the indicator.

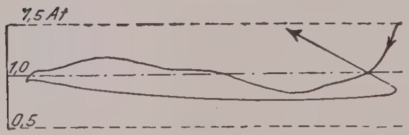

Fig. 19.

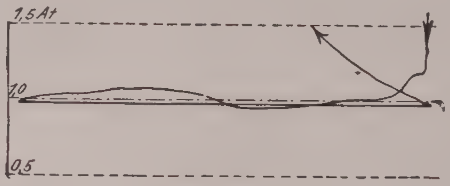

Fig. 20.

Values from Practice. The pressure along the exhaust line is usually from 15 to 16.5 lbs. abs., sometimes more; the exhaust temperature (outside of the cylinder close to the exhaust valve) varies from 950° to 1450° F. abs. As might be expected, this temperature may be raised considerably by use of rich mixtures higher (values of p_c), late ignition, strong after-burning, etc. Increased piston speed and less effective cooling on the other hand raise this temperature but little.

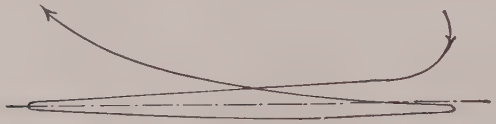

Fig. 21.

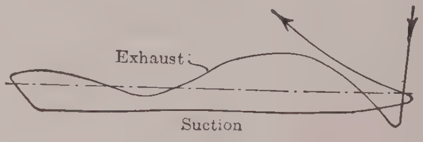

Fig. 22.

For values of the temperature T_e at the moment of exhaust opening, see p. 38.

5. The Heat Interchanges between charge and envelope during the individual strokes may best be collectively studied by means of the entropy-temperature diagram (see Appendix), an example of which has already been given on page 10 in connection with the explanation on heat losses. In Fig. 23 the entropy-temperature diagrams (based on 1 kg. of charge) for a Deutz gas engine (full heavy line), a Grob & Co's. kerosene engine (broken line), and a Diesel oil engine (full light line) have been superimposed, the numbers referring to the pressures in atmospheres at the points of location.[1] Remembering that movement to the right indicates addition of heat, and that to the left shows loss of heat, the diagram gives a clear picture of the heat interchanges during the second, third, and fourth strokes. Starting with the two explosion engines, the line *a*–*b* of the Otto engine shows a moderate loss of heat throughout the entire compression stroke; in the case of the Grob kerosene engine, on the other hand, the loss is so strong as to make the line *ab′* appear almost isothermal. (It should be noted, however, that the very small pressure ordinates of the first half of the compression can only be shown approximately, so that this part of the entropy diagram is always somewhat uncertain.) At point *b* or *b′* the explosion commences, showing strong addition of heat, as a consequence of which the curves turn upward and to the right, rising to *c* and *c′* respectively. Apparently the maximum pressure coincides with the maximum temperature in both diagrams. The expansion in the explosion engines takes place along the lines *cd* or *c′d′*, with constant further addition of heat, due to after-burning, in the gas engine, and in the kerosene engine almost adiabatically after two thirds of the expansion stroke is completed. But since some heat must be lost to the cooling water, there must also be after-burning all along the expansion line in the kerosene engine. Finally, the exhaust takes place along the lines *da* and *d′a*, during which heat is largely abstracted from the cycle.

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[1] Krauss, Z. d. Oesterr. Ingr. & Arch. Verein, 1898, No. 10. The figures in this diagram have not been transposed to English units, since they are only used for comparison.

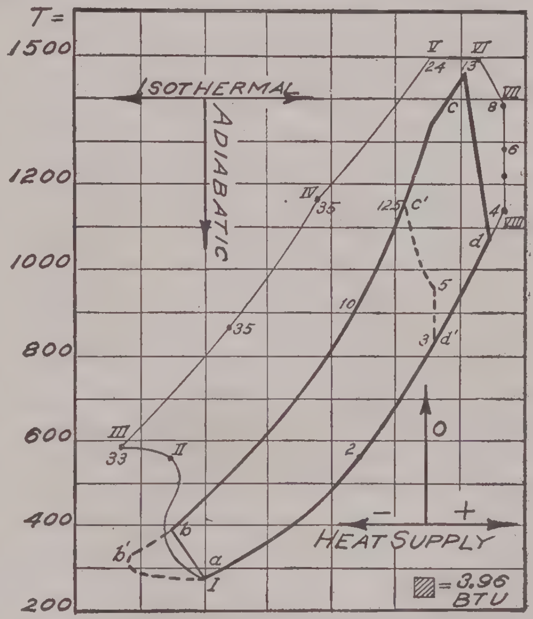

Fig. 23.—Entropy Diagrams for Constant Volume and Constant Pressure Engines.

In the entropy diagram of the Diesel engine, line I–III represents the compression stroke. The marked changes of the direction in this curve are perhaps not absolutely certain, owing to the reason mentioned above, but there is no doubt that the general trend of the line to the left shows that the compression is accompanied by a loss of heat. This loss is so marked beyond the point II that the temperature of the compressed air remains practically constant. At point III injection and combustion of the fuel commences, at once the line turns upwards and to the right, showing continued, strong supply of heat to the point V. During the same time the temperature rises rapidly from 580° to 1510° C. abs., that is, the increase is in the ratio of about 2.6 to 1. The temperature increase does not stop with the attainment of maximum pressure (point IV), as in the case of the explosion engines, but continues beyond this point to the point V, where the pressure has decreased to 24 atm. Between V and VI the temperature of the mixture remains practically constant (in this particular case); the heat used by the work of expansion and that lost due to cooling is therefore replaced by after-burning along this line. From VI to VII the heat supply from the latter cause is no longer quite able to fully replace the loss due to the other two causes, hence the temperature gradually drops, the change from VII to VIII being apparently purely adiabatic. The continued heat loss to the cooling water is therefore also in this case replaced to the end of expansion by after-burning. At the beginning of exhaust at point VIII the gases still have a temperature of over 1100° C., which, due to the heat loss in the exhaust gases, rapidly drops to about 300° C. at the point of beginning.

All three of the diagrams refer to engines brought out between 1895 and 1897. The changes which the pressure-volume diagram of the Diesel engine has experienced in the meantime would manifest themselves in the entropy diagram in that, owing to the increased supply of heat (or the increased temperature), the points IV and V, and consequently also VIII, would be moved correspondingly higher up.

In the writer's opinion, none of these entropy diagrams can claim perfect accuracy. That for the Diesel engine probably deserves the greatest credence because, except for the specific heats, complete data for its computation were given in Schröter's report on the Diesel engine of 1897. On the other hand, the lines for the diagrams of the explosion engines appear to have been determined on the basis of some assumptions at least open to questions. But in general the diagrams serve to give a picture of the heat interchanges in the cylinder sufficiently correct in its main features, which is all that is desired here.

II. The Events of the Two-stroke Cycle

In the usual form of 2-cycle engine, in which the exhaust gases are discharged during the time that the piston passes the dead center, the compression, combustion, and expansion of the charge take place exactly as in 4-cycle machines, so that as far as these events are concerned, we may refer to the previous articles. From this point of view, the only events belonging peculiarly to the two-stroke cycle are the charging and discharging actions of the mixture or scavenging pumps, which actions may continue into the cylinders or receivers. Unfortunately, as far as the development of the 2-cycle engine is concerned, but little has been done to systematically investigate and clear the most important internal phenomena connected with the charging and discharging actions of the two-stroke cycle. Inventors and manufacturers who have developed 2-cycle engines are very reticent when it comes to discussing their good or bad experiences with this type of machine; and as for scientific investigation, the two-stroke cycle no longer seems to offer to the laboratories of our technical schools any points worth clearing up. The natural result is that to-day, as for the last thirty years, the designer is compelled to solve all fundamental as well as constructive problems relating to the 2-cycle engine for himself, by the aid of trial, experience, and observation.

Among the problems open to debate regarding the 2-cycle engine might be mentioned the following:

The pressure of the scavenging agent, and the duration of the scavenging period.

The size and position of the scavenging ports.

The point of admission of the scavenging agent, whether at the cylinder head or near the piston face.

The question of intermediate receivers, whether any should be used, and if so, the proper size.

The question of excess of scavenging air, whether much or little, etc.

The following discussion is very largely based upon the experience gained by the author himself. How far the conclusions arrived at may be generalized may be seen from the appended discussion by A. Wagener regarding the charging and discharging actions in large 2-cycle engines.

1. The Pump Operations offer nothing new in themselves, since the charging or scavenging pumps do not differ materially from ordinary air pumps. The suction stroke of the 2-cycle engine differs from that of the 4-cycle only in that it is carried out in a separate pump, which latter is in all respects designed for this service only.

The large clearance space (combustion chamber of the 4-cycle engine) is eliminated, the pump cylinder is not heated up, and the valves, since they are subject to small pressures only, can be made of proper port area and of light weight. All these things combine to raise the efficiency of the pumping actions; the volumetric efficiency of the cylinder increases, which finally means that for the same piston displacement the power cylinder receives a greater quantity of cooler air than if the suction stroke had been carried out in the cylinder itself. The frictional resistances and the volumetric efficiencies of the charging pump cylinder then depend only upon the controlling conditions of the design, and their values should not be far different from those attainable in all good air pumps, say a suction pressure of from 14.1 to 14.4 lbs. absolute and a volumetric efficiency η_ε from .95 to .97. Since the most favorable pressure of the scavenging agent is probably not higher than from .75 lb. to 3 lbs. by gauge, the rise of temperature due to compression is negligible, and, in any case, it disappears during the transfer of the air to the reservoir or the power cylinder.

In the above it is of course assumed that the scavenging air or the charge is furnished by a separate pump, independent of the power cylinder. The conditions change materially when, as is sometimes done, the front end of the cylinder is used as a pump, in which case, irrespective of constructive difficulties, the pump efficiency is seriously affected by the strong heat transfer from the power cylinder to the pump cylinder. The case is still more unfavorable when, on account of considerations of low cost of machine, the enclosed crank case is made to serve as a pump. The necessarily bad shape of the crank case volume, the very large clearance spaces, and the leakage through bearings, together with the considerable heating that the charge experiences, combine to make the entire charging action much less efficient than in the case of the 4-cycle engines. This all leads to the conclusion, at least for the larger machines, that if it is intended to utilize to the fullest extent the advantages of the two-stroke cycle, only independent pumps, which may be designed as such without restriction, should be used. The use of crank case pumps is inefficient even in small engines and should be permitted only when cheapness and light weight are the primary requirements.

2. The Phenomena of Scavenging, clearing the cylinder of burned gases, are in themselves quite complex and, as already pointed out above, are by no means fully cleared up. The 2-cycle principle will stand or fall with the scavenging of the power cylinder. Efficiency, reliability, capacity—in short, everything depends upon the thoroughness with which the burned gases are driven out of the cylinder. If, as is usually intended, the cylinder volume is to be cleared out with air, the introduction of a certain amount of air in excess is indispensable. There must therefore be available a volume of scavenging air greater than the cylinder volume, because during the scavenging period some air is certain to be lost through the exhaust ports and, unless some excess air is at hand, some of the burned gases are certain to remain. In the case of independent pumps the available volume of air may be made anything, depending upon the pump dimensions; if, on the other hand, the front end of the cylinder, or even the crank case, is used to compress the air, an excess of air is not obtainable. There is in such cases likely to be a deficiency, since the otherwise possible volume is further decreased on account of large clearance spaces and heat transfer from the power piston. If it is inadvisable to use a separate pump, it then becomes necessary, by intelligent choice of duration of the scavenging period and proper design of the cylinder and valves, to obtain, with the least possible loss of air, as nearly perfect a cleaning out of the cylinder as the conditions will allow. The more or less perfect solution of this particular problem determines almost entirely the practical value of any new 2-cycle construction.

As long as the scavenging medium is led directly from the pump into the main cylinder, its proper action is seriously hampered. With pump and main cylinder cranks 180° apart, it is possible to scavenge only to the dead center; where the pump crank leads the main crank, the scavenging air must have comparatively high pressure on account of the short time available for transfer. It seems necessary therefore to interpose between pump and power cylinder a receiver of such size that, during the entire scavenging period, the air pressure can be maintained without much drop. This pressure should be very low. An abnormally high scavenging pressure not only increases the lost pump work, but, what is worse yet, interferes with a thorough driving out of the burned gases. The author, on the basis of practical experience for the last fifteen years, must emphatically state that the much favored increase in the pressure of the scavenging air entirely fails to fulfill its purpose and that a cleansing of the power cylinder both thorough and economic is possible only with air moderately compressed. Air highly compressed enters the cylinder with great velocity, and rebounding from the inner walls, causes eddy currents of such magnitude that, from the outset of the scavenging period, burned gases mix with the incoming air and thus a part of them is retained in the remaining air or the new charge. The perfect scavenging action on the other hand shows very different characteristics. The air should enter the cylinder slowly, avoiding all counter or eddy currents, and should, if possible, in the form of a solid column drive the exhaust gases in like manner ahead of itself out through the ports. This action, however, can only be obtained with low pressures and sufficiently large ports. Success in ridding the cylinder of burned gases lies entirely in carefully avoiding any breaking up of the exhaust gases and in preventing the intermixing of air and burned gas.

The allowable minimum pressure of the scavenging agent depends mainly upon the size and frictional resistance of the transfer ports and upon the time available. Neither the size of the ports nor the time available can, for very obvious reasons, be made too great. Mathematically the question may be treated as follows:[1] Let the pressure in the receiver (or pump) be p lbs. per sq.in., and the temperature be $T°$; also let the pressure in the power cylinder be p_0 lbs. per sq.in.; then the velocity of the air will in general be[2]

$$w=\phi 58\sqrt{T\left(1-\frac{p_0}{p}\right)} \text{ ft. per sec.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (67)$$

Then, if the area of the ports equals f sq.ft., the volume of air transferred per second will be

$$V=\alpha w f \text{ cu.ft.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (68)$$

In these equations, ϕ represents a velocity coefficient and α the coefficient of contraction; the product of these two factors is the so-called coefficient of efflux $\mu=\alpha\phi$. In practice it has been found that ϕ varies from .85 to .95. The value of α cannot be definitely given since it depends upon attendant conditions, and it should therefore be determined for every case. It is safe, however, in most cases to make α approximately equal to from .6 to .65.

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[1] A comprehensive mathematical investigation of the dynamic action during scavenging and charging has been published by A. Wagener in the Gasmotorentechnik, 1903.

[2] Hütte, ed. 19, Vol. I, p. 332.

Example. Assume $\phi=.90$, $\alpha=.62$, $T=461+81=542°$, $p=16.3$ lbs., and $p_0=14.9$ lbs. Then

$$w=.9\times58\sqrt{542\left(1-\frac{14.9}{16.3}\right)}=367 \text{ ft. per sec.},$$

and $V=.62\times367f=227f$ cu.ft. per sec.

It will be seen from this that even with small pressure differences the velocity of flow is considerable.

The author's requirement calling for "lowest possible pressure of the scavenging air" is not accepted without qualification by Mr. A. Wagener, who for about ten years has been engaged in studying the dynamic actions in the cylinder of the Oechelhäuser engine. The reasons for this dissent are given in the following in Mr. Wagener's own words:[1]

"H. Güldner, in his "Entwerfen und Berechnen der Verbrennungsmotoren," concerning air storage receivers, voices the opinion that it is desirable to work during the period of scavenging with as small a pressure drop as possible and hence to make the receivers as large as possible. This requirement seems quite justified for certain types of engines; but to generalize it and to make it apply without qualification to all 2-cycle engines would be a mistake, for the reason that for certain 2-cycle engines it is contradictory to another requirement of fully as great importance. Such engines are those utilizing the lean industrial gases, in which the air for both scavenging and charging comes from the same receiver and flows through the same transfer ports, and whose speed regulation must be within very close limits. When a rich fuel is used in an engine, the amount of such fuel required per cycle even for maximum load in large engines is comparatively small. It is quite possible to force this amount of gas into the cylinder without simultaneous addition of any considerable quantity of air and still obtain a sufficiently intimate mixing of the fuel with the volume of air already present in the cylinder. In engines of the Oechelhäuser and Junkers type, for instance, the charging pump was made to force a certain amount of illuminating gas to which only a small amount of air had been added into the cylinder during the compression stroke. Under these conditions the combustion line testified to the formation of a very good fuel mixture. Under low loads on the engine it was even possible to force fuel gas alone into the power cylinder with equally satisfactory results. If, on the other hand, lean gases are to be used, the gas volume required for each working cycle is, in general, so large that a method of operation like that above outlined is entirely unsuited. What would happen is that only at the beginning of the gas-charging period would there be any mixing of the incoming gas and air, for the rest of the time the incoming gas column would merely displace the air. With lean gases, therefore, it becomes necessary to form the fuel mixture in the charging pump or the proper quantity of air must be injected into the power cylinder at the same time with the gas, so that the mixture may form during the period of introduction. Experience has shown that there are serious objections to the first of these schemes, that is, to form the mixture in the charging pump, one of which is that the receivers and passages are filled with an explosive mixture. If under these conditions, owing to the formation of mixtures deficient in air—a state of affairs which may be, but is not always, prevented, especially at starting—the charge should be ignited during the transfer period, the result would be the explosion of the mixture in the receiver, which accident nearly always results in damage to some part of the engine and especially to the charging-pump valves.

If the mixture is formed by simultaneous introduction of gas and air, it becomes necessary, when the amount of fuel used is changed for purposes of speed regulation, to correspondingly change the amount of air if the combustion is to remain the most efficient possible. The governing arrangements, by which the problem of measuring the required quantities of gas and air must be carried out, may be constructed with the most simple means, and with promise of greatest reliability, when, in order to control the volumes of the gases transferred, the pressures in the receivers are suitably changed. The most economic way of changing the pressures is by decreasing the volumes of the gases delivered by the pumps, so that with a decrease in pump capacity there shall be a corresponding decrease in the pump work. The simplest means of obtaining this result is probably to arrange a by-pass valve under the control of the governor, so that at the beginning of the discharge stroke a larger or smaller quantity of gas or air, depending upon the position of the valve, is forced back in the suction mains at practically atmospheric pressure.

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[1] From a lecture delivered before the Bavarian District Society of German Engineers, 1903. This lecture with some elaborations was published in the Gasmotorentechnik, and deserves attention on account of its mathematical treatment of the charging and discharging actions of 2-cycle engines.

Now, however, if the change in the quantity of gas transferred is to be brought about by a change in the pressure in the receivers, it becomes desirable to make the volumes of the latter, including that for air, as small as possible. The reason for this obviously is that, the greater the volume of these receivers, the greater will be the time elapsed between the moment that the governor begins to affect the gas volumes delivered by the pump and the moment that the volumes delivered from the receiver begin to be affected in the corresponding way.

A drop in pressure of say 1.5 lbs. would call for a receiver volume equal to approximately 210 cu.ft. With such volumes, governing arrangements as above outlined would fail completely; and the question therefore arises whether the requirements that the air receiver should be made as large as possible, is of enough importance to warrant seriously curtailing freedom of choice regarding the details of construction.

In Oechelhäuser machines results satisfactory in every respect have been obtained with dimensions of the gas and air receivers such that, for the maximum gas volumes transferred, the pressure drop is as high as 14 lbs. per sq.in. This construction calls for comparatively high pressures at the beginning of the scavenging period, up to 22 or 23 lbs. per sq.in. absolute for high speed engines. In spite of this fact the difficulties mentioned in Güldner's book have not manifested themselves up to this time in Oechelhäuser engines. It is of course quite possible that the ring of ports used for the air inlet in this machine is specially suited for such conditions of operation. Güldner says, among other things, concerning this question: "Air highly compressed enters the cylinder with great velocity, and rebounding from the inner walls, causes eddy currents of such magnitude that, from the outset of the scavenging period, the burned gases mix with the incoming air and thus a part of them is retained in the remaining air or the new charge." This is no doubt practically what happens when the air is introduced from one side of the cylinder only, as for instance, when the ports are confined to only a part of the circumference. If, on the other hand, the inlet ports are arranged completely around the circumference, the writer is of the opinion that, in view of the initial guidance that the air currents receive when they strike the face of the piston and their subsequent mutual interference, the final result is the formation of a column of air of sufficient solidity to clear out the cylinder. In practice, the minimum volume of scavenging air required to obtain reliable operation serves in every case as a criterion for the efficiency of the scavenging action. Whether the introduction of this minimum quantity of scavenging air is always combined with the greatest economy of operation is altogether another question.

It might finally be mentioned that the magnitude of the pressure drop has but very little influence upon the work required by the pumps. If, for instance, in any given case the pressure drop is 8.5 lbs. per sq.in., the charging pump must of course compress to about 23 lbs. per sq.in., which is comparatively high; but the transfer from the pump to the receiver commences almost with the beginning of the compression stroke, since, owing to the previous operation, the pressure in the receiver is practically atmospheric. On the other hand, if the pressure drop is say 1.5 lbs., the pump has to compress to only about 19 lbs.; but since during the previous operation the receiver pressure has dropped to only 17.5 lbs., the pump must first compress to this pressure before the transfer to the receiver can begin. Comparing the indicator cards from two pumps operating under these conditions, it will be found that the work required in the former case is but little greater than that in the latter. The most important factor determining the work required by the pump is the value of the mean receiver pressure p_m, but the latter has no direct relation to the volume of the receiver."

Another requirement for good scavenging is that the air be not admitted until the exhaust is complete, that is, until pressure equilibrium in the cylinder has been established. An earlier admission of air is possible only with unnecessarily high air pressure, the use of which, as already shown, is bad in itself and should be avoided. The burned gases are still under high pressure and consequently their temperature is higher than ordinary, which tends to decrease the weight of air or new charge in the cylinder owing to the heating they receive.

It is quite difficult to attain a complete pressure drop to atmosphere, or nearly so, with the ordinary exhaust valve in the short time available in most cases. A ring of ports in the cylinder wall is much better suited to the purpose, and consequently much used in the design of 2-cycle engines. But even in this case it is best not to admit the scavenging air until the dead center, to make certain under any circumstances that the pressure drop is complete and that the exhaust gases are cooled as far as possible. With the piston controlling the exhaust ports, there will then still be available a period of time for scavenging and charging equal to the time required by the piston to cover 10–12% of its stroke at that end of the cylinder. Better results will thus be attained than if the air had been forced into the as yet highly compressed and heated gases at an earlier period. As compared with 4-cycle machines, the beginning of compression in 2-cycle machines may commence a little later on the return stroke without much harm, because in the former the compression must commence with a pressure p_a invariably somewhat below atmosphere, while in the 2-cycle engine the compression commences with a pressure of from .5 to 1.0 lb. per sq.in. above atmosphere (see p. 52).

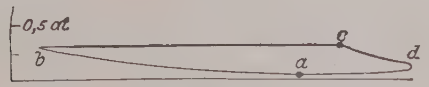

FIG. 24.

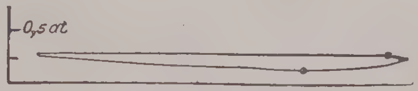

FIG. 25.

It is desirable to have the pressure drop in the air receiver as small as possible, in order that the scavenging may continue with undiminished force to the end of the period. The best means of attaining this end is to make the receiver as large as possible and to commence the transfer of air not too early in the stroke. The two diagrams of Figs. 24 and 25 make the points clear. The first is taken from a receiver of too small volume. The pump delivers air into the receiver from a to b. From b to c the pressure remains constant at about .3 *atm.* (4.2 *lbs. per sq.in.*) *above atmosphere.* Scavenging commences at c, but at the outer dead center d the pressure has already dropped to .1 atm. (1.4 lbs. per sq.in.) and beyond this point the drop is very slow. It is evident that the transfer of air had already ceased before the inlet valve closed at a; that is the best part of the time available for scavenging has not been used. After a considerable increase in the volume of the receiver, diagram Fig. 25 resulted. The maximum scavenging pressure is now a little less than before; the transfer commences a little later, but continues at about the same rate until the closure of the inlet valve, since in this case the receiver pressure does not drop below .2 *atm.* (2.8 *lbs per sq.in.*) *above atmosphere.* The pressure at the beginning of compression in the power cylinder is correspondingly higher, and since on account of somewhat later introduction the air is less highly pre-heated, there will be a greater charge volume in the cylinder, which finally means greater engine capacity.

The manner of scavenging without the use of intermediate receivers is well illustrated in Figs. 26 and 27, the diagrams being taken from a crank-case 2-cycle engine of the author's own design of 1895. The crank-case pump draws air from a to b, Fig. 26, and compresses it into the large clearance spaces up to the point c, approximately to .3 *atm.* (4.2 *lbs. per sq.in.*). After the burned gases have been exhausted from the cylinder, the transfer of air proceeds with rapid drop of pressure. Scavenging is completed at the point *d*, because the piston then commences its return stroke, and the little overpressure still remaining in the crank case suddenly disappears. In order to make the scavenging period independent of the movement of the piston, the transfer pipe between crank case and cylinder was made as large as possible, and was separated from the crank case by a flap check valve. In this way the pipe was converted into a receiver, although of insufficient capacity. The diagram obtained, Fig. 27, sufficiently illustrates the improvements thus made in the scavenging process. The check valve admits the air to the transfer pipe, but prevents its return, and thus permits a continuation of the scavenging action even after the crank-case pump has started on its suction stroke. This is about the only satisfactory way by which the burned gases can be removed from the cylinder in the case of engines using the crank case as a pump.

The rapid and complete drop in the pressure of the burned gases to near atmosphere after the exhaust valve has opened, is of extreme importance in case the fuel mixture instead of air is used for scavenging. Even in this case a little air is usually introduced ahead of the mixture, but this does not in itself offer safety. It is even possible that, owing to this introduction, some of the unburned gas particles contained in the burned gases may be ignited by the fresh air, thus firing the entire incoming charge upon entrance. *The only safeguard against an occurrence of this kind is to hold back the scavenging medium until the exhaust of the burned gases is complete.*

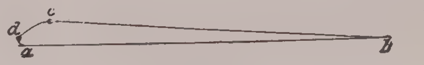

FIG. 26.

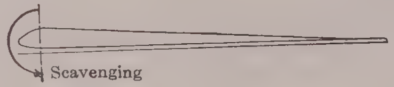

FIG. 27.

The difficulties that may be encountered through neglect of the above rule are well indicated by Mr. A. Wagener in a lecture delivered before the Society of German Engineers at their general convention in 1900. Mr. Wagener tells of some of the experiences encountered in putting into operation a 600 H.P. Oechelhäuser and Junkers blast-furnace gas engine as follows:[1]

> "An attempt to operate the machine at normal speed failed because, even after a comparatively short period of operation at less than normal load, the working cylinders became so hot that the occurrence of pre-ignition was feared, and the latter actually did occur from time to time in the right-hand cylinder, as during the tests this machine probably carried somewhat more load than the other half of the unit.
>
> The trouble in this case was found to be caused by the fact that the exhaust ports did not open early enough, and that on the opening of the air ports a part of the burned gases escaped through these into the air jacket surrounding the air ports. This action could be traced at a speed as low as 120 r.p.m., although the interference was not strong enough at this point to cause serious trouble.
>
> The inter-related actions of air and burned gas are very clearly shown in Fig. 28. The upper curve (1) shows the pressure of the burned gases in the cylinder during exhaust. The line was taken by means of a weak spring, and transferring it to the drafting-board the pressure ordinates were further enlarged with the greatest care. At the same time that line (1) was obtained, a pressure diagram (2) was obtained from the air jacket surrounding the air ports, and this line was drawn over (1) to the same scale. In obtaining this line, the reducing motion of the indicator was set parallel with the outside cranks, so that the abscissas represent the

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[1] Z. d. V. D. Ing., 1900, p. 1517.

positions of the back piston. Fig. 28 also shows the relative proportions and positions of the air, mixture, and exhaust ports. The exhaust ports are of course controlled by the other piston, but have been transferred in the figure with due regard to the relation between stroke and port opening. In this way, if an ordinate be drawn at any point corresponding to a given piston position, that part of the areas not cross-hatched in the figure, lying to the right of this ordinate, will represent the port opening for each ring of ports at that piston position.

The exhaust ports open at *c*, but at first the pressure drop is not very marked. This is probably partly due to friction in the passages, and also to the fact that in any case a certain amount of time is required to produce the necessary acceleration even in the comparatively small mass of the burned gases. Further, there is some lag in the propagation of the pressure drop from the neighborhood of the exhaust ports to the middle of the cylinder where the indicator is located. The same interesting phenomenon occurs in the pressure diagram for the air jacket (2). The air ports start to open at *a*, at a point where the pressure in the cylinder is still 47.3 lbs. per sq.in. abs. One would assume that the immediate action would be for the exhaust gases to rush into the air jacket, raising the pressure in the latter considerably. It will be seen, however, from the diagram that this occurs at a point considerably later in the stroke. At the point *d* the pressure in the cylinder is the same as that in the air jacket, and it would naturally be expected that from here on both lines should indicate an approximately similar pressure drop. The diagram, on the other hand, shows a further increase of pressure in the air jacket, and a continued drop in the cylinder.

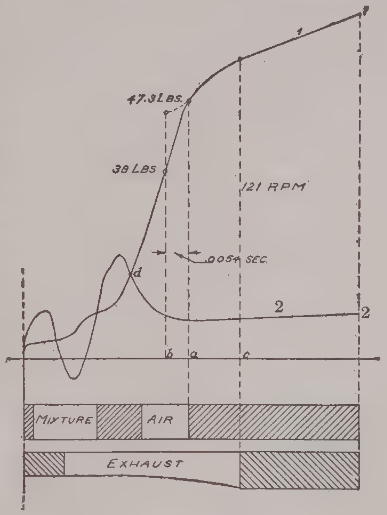

FIG. 28.

The further path of curve (2) shows that soon after the pressure in the air jacket attains its maximum value, the mixture, consisting of burned gases and air, rushes into the cylinder with such high velocity that a vacuum is formed in the air jacket itself. This in turn is followed by a rise in pressure due to the air flowing in from the main receiver. Near the dead center, therefore, very complicated inter-actions occur in a very short space of time, and the masses of gas and air are set into violent vibration, a phenomenon to which the writer will again refer later on.

The diagrams of course only show pressure variations in the vicinity of the indicator, and it should not be forgotten that the lines are apt to be somewhat modified by the inertia of the moving parts of the indicators themselves.

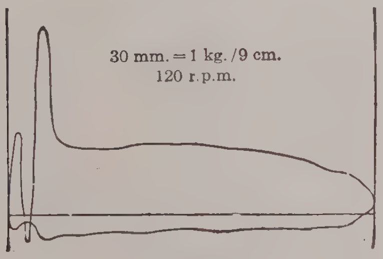

FIG. 29.

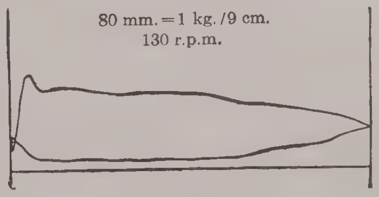

FIG. 30.

It is very interesting to determine the rapidity with which the actions described occur. From the moment of opening of the air ports, for instance, to the moment that the exhaust gases commence to enter the air jacket with rapidity, that is, from *a* to *b*, the time interval is .0054 sec., during which the pressure in the cylinder drops from 47.3 lbs. abs to 38.0 lbs. abs. If it be assumed that the pressure drop is uniform throughout the volume of the cylinder, it follows that in this short time 54 ltr. (1.906 cu.ft.) of burned gases escape through the ports. The mean port area is 257 sq.cm. (42.7 sq.in.), and hence, if the gases flow through the ports at approximately atmospheric pressure, the mean velocity would have to be about 365 m. per sec. (1197 ft. per sec.). It is of course nearly impossible to determine the absolute velocity, since no reliable means are at hand for measuring the pressure differences between the inlet to and exit from the ports, but the above computation, even if very approximate, shows that the maximum velocity of the outflowing burned gases reaches an extraordinarily high value.

The difficulty above outlined was remedied in the machine under discussion by elongating the exhaust ports toward the middle of the cylinder, a distance of 20 mm. (.8″).

Fig. 29 shows the complete diagram from the air jacket as taken before changing the exhaust ports, while Fig. 30 shows the diagram as at present obtained. It will be seen that the sudden pressure drop and the violent vibrations of gas and air are almost entirely eliminated. Further, no vacuum is formed at any time between the air ports and the controlling valve on the air receiver, and the entire operation of transferring the air takes place under much quieter and more efficient conditions.

The machine now operates without difficulty at 130 to 135 r.p.m., and, even under overloads, the cylinders heat up to so slight an extent that the hand may be laid for a few moments upon the uncooled outer ends of the cylinder without discomfort.

III. Comparison between 4= and 2=cycle Engines

1. Considered from the standpoint of **theoretical thermal efficiency,** both principles of operation possess exactly the same value, their working cycles do not differ in any respect. The **real thermal efficiency,** however, is affected by a series of practical conditions, several of which speak very strongly *for the* 2-*cycle engine.* Since the specific capacity of the 2-cycle engine is twice that of a 4-cycle machine, it follows that the former requires only one-half of the cylinder volume of the latter. The comparatively greater superficial area of the smaller cylinder volume is in favor of the 2-cycle engine in so far as relatively more heat of compression is transferred to the cooling water, permitting the use of higher compression pressures. Besides possessing practical advantages, the latter fact means a better thermal efficiency. It is a fact of course that the greater heat transfer, on account of relatively greater superficial area, also exists during combustion as well as compression. This is not a desirable condition, but it seems that the disadvantage is about balanced by the fact that the average combustion temperature of lean mixtures highly compressed is correspondingly less. It has been found that the losses to cooling water in the two types of engine are not far different, in fact this loss sometimes shows less in 2-cycle machines than in 4-cycle machines of like capacity. A 15 H.P. Bénier 2-cycle engine, for instance, tested by Witz, required 61 lbs. of cooling water per B.H.P. hour, while an Otto engine of the same capacity, examined by Köhler, required 54 lbs., both being operated with producer gas. A 600 B.H.P. Simplex 4-cycle blast-furnace gas engine, according to Hubert's tests, required as much as 110 to 143 lbs. of jacket water per B.H.P. hour. From the author's own experiments, small 2-cycle engines lose to the cooling water from 3400 to 3800 B.T.U. per I.H.P. hour, the larger sizes from 3000 to 3300 B.T.U. As in the case of 4-cycle machines, this amounts to from 35–40% of the heat supplied.

Fear might be expressed that the scavenging air may become strongly preheated by the burned gases to be displaced. This however does not take place if the scavenging is properly done, i.e., if the transfer of air does not commence too soon. The temperature T_a of the charge at the beginning of compression in 2-cycle machines is usually not higher, rather lower, than in 4-cycle engines.[1] This fact may be easily explained by considering that the transfer of air, a process in itself combined with a temperature drop, takes place with great rapidity, taking up from about $\frac{1}{10}$ to $\frac{1}{8}$ of the time of one stroke, while in the case of the 4-cycle the charging action occupies the time of one entrie stroke, hence the heat absorption by the charge must be much more marked. A rise in the temperature of the charge after the closure of the transfer ports no longer effects the engine capacity. The latter is increased by the fact that, in 2-cycle engines, compression commences when the charge is at a pressure already slightly above atmosphere, hence the charge weight is greater than if the charge had been drawn into the cylinder by suction. Assuming that at the end of the scavenging period the pressure in the cylinder of the 2-cycle engine is say 1.05 atm., while the pressure at the end of the suction stroke in the 4-cycle cylinder is say .9 atm.; then, irrespective of any influence that the charge temperature may have, the 2-cycle cylinder contains $\frac{1.05-.9}{.9}\times 100=16\%$ more charge weight than the 4-cycle cylinder, and has hence gained the same amount in specific capacity. (For the reason that the external useful work is, other conditions being the same, directly proportional to the charge weight.)

2. The second important basis of comparison of the two engine types is found in the **friction losses of the machine.** In this respect the 2-cycle engine is under a disadvantage as compared with the 4-cycle in that its charging and discharging actions require a greater movement of the volumes of the gases involved, since the charge must pass through two cylinders. All other conditions being the same, this fact would require a greater amount of work, and this is what is usually found in 2-cycle engines as constructed to-day. It should, however, be noted that a charging pump specially constructed for its work can operate at a higher efficiency as a pump than the main power cylinder, and also that in the 2-cycle engine the entire exhaust stroke is eliminated, i.e., the resistance p_r does not exist. Estimating the suction pressure in the 4-cycle diagram at $-.1$ atm., the resistance during exhaust at $+.1$ atm., at the same time taking the vacuum during the suction stroke of the 2-cycle pump equal to .03 atm. (which should be easily reached), the gain in the 2-cycle engine would be $.1+.1-.03=.17$ atm., and this amount, with skilful and intelligent design, should be sufficient to cover the extra work required in the transfer of the charge in the 2-cycle engine. That the moving of greater volumes of gas may not represent an increased loss of work is quite clearly shown in some tests by Humphrey on a Crossley and a Premier engine, both large horizontal 4-cycle machines (see Part III). In spite of the fact that in the Premier engine the cylinder was very thoroughly scavenged (air excess 250%) the work done in charging and discharging at the greatest total load of 650 I.H.P. was only 5.5%, while the same work in the Crossley engine, of appoximately the same capacity, represented 7.3% of the total load.[2] Consequently, although at present the pump work in 2-cycle engines is still from 8–10% of the I.H.P., as compared with 6–7% in 4-cycle engines, it does not necessarily follow that this defect is permanent; the further development of 2-cycle types will undoubtedly yet wipe out the 2 or 3% of difference still existing.

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[1] The comparatively stronger heat transfer during compression and the lower temperature T_a at the beginning of compression in 2-cycle machines made themselves strongly felt in the first Diesel-Güldner 2-cycle engine by the fact that the compression had to be higher than in 4-cycle engines by from 60 to 90 lbs. in order to produce an end temperature T_c sufficiently high to insure regularity of ignition.

[2] See *Engineering*, 1901, p. 197.

The greater part of the friction loss of the machine is found in the rubbing parts. With regard to this no definite comparison can be made, since no practical data is available, and mathematical determinations of the friction losses based on arbitrary assumption would be worse than useless. Some light may, however, be obtained from a few general considerations.

According to an old reliable rule of thumb, the friction loss in similar piston engines is the greater the greater the volume of piston displacement per horse-power. According to this rule, 4-cycle and 2-cycle engines would then be on an equal footing if the volume of the pump be added to the cylinder volume. When it is considered, however, that the 2-cycle pump piston works against very low pressure only, that therefore it may be fitted with very light rings and be an easy fit in the bore, and further that this light piston operates all the time on cool, well lubricated surfaces, there can be little doubt as to which cycle possesses least pump friction. When a piston, built for operation against pressures from 300 to 400 lbs. per sq.in., is for one-half of the time called upon to operate against pressures of from 3–4 lbs. per sq.in., i.e., pressures averaging only 1% of those for which the machine member was designed, there must evidently be useless sacrifice of mechanical efficiency and of economy in material. Such, however, is exactly the condition of things in the 4-cycle engine.

It is not at all necessary that the scavenging pump should be an independent machine, an added complication requiring a certain amount of work for its operation. It is possible, for instance, to so design the cross-head as to make it act as an air pump, as was done in the Diesel-Güldner engine. This construction secures the added advantage that the power piston and cylinder are relieved of the side thrust of the connecting rod. (It is pointed out in Part III that to make the power piston a machine member, already fully loaded by the explosion pressure and working on highly heated surfaces, also carry considerable side thrust, is fundamentally a mistake.) The adoption of such special designs makes it possible to construct an engine with pump, in which the piston and guide friction is not only not greater, but even smaller, than in an engine without the pump. That this can be done is shown by the example of the Bénier engine already quoted, this engine with scavenging pump showing a mechanical efficiency of 81% at only two thirds load. The 500 H.P. Premier engine above mentioned showed a mechanical efficiency of 88.8% without the pump, and 83.8%, allowing for the pump work, while the Crossley engine quoted at maximum load showed a mechanical efficiency of only 83.0%.

As compared with the work of charging and discharging and the friction in pistons and cross-heads, the rest of the friction losses are of secondary importance. It need only be briefly pointed out that, since the piston of the 2-cycle engine has approximately only one-half the area of the 4-cycle piston for the same capacity, the crank mechanism and shaft may be made correspondingly smaller, and the friction loss in pins and bearings is correspondingly less. The latter is further decreased by the fact that the fly-wheel of the 2-cycle engine is lighter by approximately 50%. This is corroborated by the common practical experience that the mechanical efficiency of a given engine drops a few per cent, when an extra heavy wheel (for electric light service) is substituted for the ordinary wheel.

3. From the standpoint of design, the relation between the 4-cycle and the 2-cycle engine has already been made sufficiently clear in the preceding articles. The superiority of the latter, as regards regularity of operation, space required, weight, and cost of construction, is patent in all cases where the proper design has been employed. In the building of large engines, the complexity of the 4-cycle construction more strongly emphasizes the advantages of the 2-cycle, especially so the nearer the sizes approach the limit of capacity. For the large machines the main issue of the "cycle" controversy is no longer the fuel consumption, but, very simply, the possibility of construction under ordinary manufacturing conditions, and of reliability of operation. The fact that beyond a certain size the 4-cycle engine fails to meet these conditions in the best way, has served to clearly show the advantages of the 2-cycle machine, and has done much to give it its new lease of life. Even Ernst Körting, the successful builder of 4-cycle engines, agrees with the writer in this, stating on one occasion[1] that "the successful construction of 2-cycle engines, either single- or double-acting, is almost a necessity in the development of large units, such as required in the operation of steel works. The weight of the various machine parts and their dimensions increase so enormously, when it is attempted to generate these powers in one 4-cycle cylinder, and, further, the growing complexity of the machine and the unreliability of operation become so serious when it is attempted to get the same power by the combination of four single-acting 4-cycle cylinders, that either type of construction can only be *considered as a make-shift until an efficient and reliable 2-cycle machine appears on the market.*"

Practically the same conditions exist to-day, although the builders of large 4-cycle engines have again taken up the construction of double-acting engines, thus doubling the specific horse-power capacity. (In this connection it should be noted that this method does not increase the capacity of the cycle itself, but, by making a given cylinder double-acting, merely raises the capacity of that cylinder.) A double-acting 4-cycle machine is on the same footing regarding capacity as a single-acting 2-cycle engine; but since, previous to the construction of these double-acting 4-cycle engines, large 2-cycle engines had already been built, the relative positions of the two types of machines are not changed in the least. Concerning the question of regulation, i.e., the uniformity of tangential effort at the crank, the adoption of the double-acting principle in the 4-cycle improves things but little as compared to the 2-cycle. The regulation of the double-acting 4-cycle engine compared to that of the single-acting 2-cycle engine is still in the ratio of 2 to 3, in favor of the latter machine, and in order to create anything like equality in this respect when the 2-cycle is made double-acting, two double-acting 4-cycle cylinders would have to be used.

4. Concerning the question of economy, there is little more to be said, after pointing out the practical equality of the two principles of operation regarding the thermal efficiency and friction losses. It can easily be seen from tests on the 2-cycle engines of Bénier, Güldner, and Diesel-Güldner that in every case the indicated thermal efficiency η_i is very satisfactory, and that, even in cases where the machine is unduly handicapped by friction losses in the pumps, the economic efficiency η_w in 2-cycle machines is but little inferior to that found in good 4-cycle engines. Considering the question of economic operation in its entirety, i.e., not only considering fuel economy but also first cost, maintenance, reliability, etc., any 2-cycle engine of fair grade but of absolute reliability will be quite able to take care of itself.

The requirement laid down by Prof. Riedler, that for best results in the building and operation of large gas engines the cycle should be the two-stroke cycle with capacities up to 500 H.P. in the cylinder,[1] at first seemed to many to be impossible of realization. But the question has in the meantime grown in economic importance and in a short time the final solution will most likely not depend upon what *can* be done, but upon what *must* be done.

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[1] Z. d. V. D. I., 1902, p. 127.

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[1] This is Riedler's opinion as laid down in "Schnellbetrieb" (Part XI, p. 37), three years ago. Since that time Prof. Riedler has developed into a very strong advocate of the 4-cycle engine, in fact, of a certain double-acting 4-cycle engine. When the author in spite of this adheres to Prof. Riedler's early favorable opinion of the 2-cycle principle, he does this because it seems to him that in the meantime conditions affecting the problem have not changed in the least, and also because facts there cited by Prof. Riedler in his usual clear manner seem to him to point much more strongly to the conclusion above quoted than to the diametrically opposite view expressed before the 45th General Convention of the Society of German Engineers.

The expression coined by Prof. Riedler on this occasion, that

> "the gas engine will revert to its
> original starting point, that is to the
> 4-cycle principle,"

certainly does not offer any serious obstacle to the further development of the 2-cycle engine. It will probably temporarily aid certain 4-cycle constructions—it will not, however, serve to maintain the 4-cycle principle in the large engine field. Qui vivra verra!

PART II

THE DESIGN AND CONSTRUCTION OF INTERNAL-COMBUSTION MOTORS

A. FUNDAMENTAL CONSIDERATIONS

I. Less "Invention," more "Rational" Design

The most serious handicap to the development of any type of machine is an aimless multiplicity of its various constructed forms. These many forms, not called forth by any real need but mostly through the ambitions of restless inventors, lead to designs which, in nine cases out of ten, are at variance with even the most fundamental principles of construction. And although such performances soon find their proper reward in brevity of life, there still remains this general disadvantage, that the endless procession of such one-day constructions has a deterrent effect upon sane efforts in the branch of industry concerned, and that all constructions, whether capable of life or not, are weighed down with the same appearance of unreliability.

This abnormal tendency to invention was especially pronounced in the case of gas-engine construction. There was a time—and it is not even now entirely passed—during which the employment of known and tried devices was frowned upon, when not only the uninitiated but also the professional men were strongly attracted by what seemed new or unknown, and when the happy inventor of a hot tube or an exhaust muffler could at a turn of the wrist become the founder and director of a gas-engine works. It went without saying that every "independent" designer, besides being conversant with the principles of ordinary design, had to have a new valve gear or a new method of regulation of his own, and it has actually happened that one of these inventive geniuses (?) designed a fly-wheel to act at the same time as a tank for cooling water, merely to show something "never before offered." No wonder, therefore, that the practical value of most of the new engine systems was in the inverse ratio to the number of patents covering them.

If anything has contributed more than any other cause to the internal soundness of our present day gas-engine practice, it is the fact that this reckless inventing, this search after new things, is gradually being abandoned. In its place we find business-like design and development of those types which have managed to maintain themselves through the disorder of earlier years, and which have become, to a greater or less degree, fundamental forms. For the gas-engine designer of to-day the most important problem is to develop these fundamental forms with the aid of the well-known elements of general machine design, and upon the basis of principles proven by science and tested by practice. In the solution of this problem, the less free rein is given to the inventive bent, the better for the designer and the problem. This applies especially to the younger designer whose ambition is very apt to lead him to look upon inventing as a necessary preliminary to, or even as the ultimate aim of, his labors. In doing so he loses sight of the real aim of every sane engineering problem, i.e., to furnish something which embodies the most advanced and best ideas, and which shall be of practical utility. We do not design in order to invent: on the contrary, for important engineering work it is usually one of the requirements to its successful accomplishment that it be carried out only on known and tried ground. With a great deal of reason, therefore, a good designer generally has the name of being an unsuccessful inventor—and *vice versa.*

It is of course not in the least the intention to underrate the importance of inventions which are based upon the recognition of an actual need, and which have their origin in true mental labor; but how little of this finds expression in the ordinary run of patents covering designs and constructions!

Here as everywhere in first place comes the alpha and the omega of every theory of design, which is *the proper adaptation of the constructive means at hand to the existing conditions.*

That which is unquestionably the right thing in one case may be entirely out of place in another. A small engine must be designed from an entirely different standpoint than is proper for a large industrial machine; a marine engine calls for a construction differing from that of a stationary motor; and an originally stationary oil engine is by no means a locomobile, because it is placed upon four wheels. Only in one sense is there for all forms and types the same requirement, that is, reliability and durability, and to it all other requirements must be subordinated. A single hour of stoppage in operation may serve to wipe out the saving in fuel for an entire year, and an engine showing 50% thermal efficiency would be a costly prime mover, if after a few years it should have to be relegated to the scrap pile.

In *general,* the following points should be kept in mind: In the case of the small engine, greatest possible simplicity, low cost, and durability; for the large engine, greatest economy, and for the automobile machine, smallest possible weight. Compared with the primary requirement, as determined in any given case, the rest are of secondary importance, or do not enter the problem at all. Of course, in the case of the small engine, fuel economy should be considered, just as we should take simplicity of construction into account in the case of the large machine, but the primary requisite should not be lost sight of in each case. There is no sense in doubling the number of machine parts and the manufacturing cost of a commercial machine to gain a saving of a few cubic feet of gas; but it is just as unwise to hesitate over the addition of another valve, or of a few pounds of metal, when the reliability and the economy of the machine can be increased thereby.

The working stresses in internal-combustion engines are from two to three times as high as those found in steam-engine practice. Besides this these stresses cannot be as certainly controlled in their time of occurrence as can those of a steam engine, and it may occasionally happen, due to a combination of circumstances, that they are changed in a manner detrimental to the entire construction. This fact makes it imperative that the designer choose only such forms in which all maximum stresses occurring are taken up as centrally as possible; which admit of definite strength computations and which preclude any hidden defects in manufacture. Plain forms are not only statically the safest and cheapest as regards manufacturing costs, but also pleasing to the technical sense if properly carried out. A tendency to excessive economy of metal, however, is apt to degenerate this plainness of form into mere ugliness, and this should be guarded against. It is for instance out of place to give the frame of a machine the shape of a plain box with beveled edges to save the cost of a better frame pattern, as is done in many English and American machines. Since every frame pattern is used a large number of times, its cost cannot seriously add to the manufacturing cost of a machine.

The good features of even the best design are not well brought out when the arrangement of the auxiliaries and the painting of the machine is out of keeping with the rest. An engine which, against a bright-red background as a body color, shows a sample assortment of nickeled parts, highly polished brass, and artistically scraped cast-iron surfaces, etc., may be an attractive exhibit for the masses, but to the professional man its appearance is repellant. The finished machine should show two colors only: a dull black or, at least, dark color, for the cast surfaces and the metallic white for the machined parts. The nickeling of entire machine parts is technically wrong and perhaps allowable only in the case of small auxiliary parts of brass, bronze, or copper. The external surfaces of machined parts need only be neatly polished or ground; high polish is out of place in machine construction, and makes an unnatural impression even in engines "on parade." The more perfect the design, the better the materials and the more accurate the construction, the less need is there for any embellishment—true worth speaks for itself.

In the general arrangement of the installation the qualifications and knowledge of the attendants must not be left out of account. It is best not to form too high an estimate of them, even in the case of small engines. An attempt to explain a complicated installation by a lengthy set of operating directions invariably fails in its purpose; the directions are either not read, or, if read, are wrongly interpreted. But few machine parts and fewer motions on the part of the attendant should be required to start and stop the machine and to keep it in regular operation. Single parts, which during the assembling of the machines are accurately set, and especially such parts in which change in position by inexperienced hands can become dangerous to the engine, had better be permanently fastened to place them out of the reach of attendants anxious to investigate or to improve.

Another point which should receive the proper attention of the designer is the item of repair, by which is meant the readiness with which worn or broken parts can be repaired or replaced by the ordinary machinist. In this respect it is of course not necessary to go as far as a certain landowner who expected that a serviceable oil-locomobile should be "capable of repair with a hoe"; but those parts at least which are sure to wear, as for instance, bearings, hot tubes, packings, studs, and bolts, etc., should be so designed that in case of need they can be renewed in every good machine shop. That important parts, like pistons and valves, should be replaced by the factory only, is self-evident.

Finally, the manufacturing cost of every new construction must not be forgotten, because this determines the sale price, which in turn affects the marketability and the commercial economy of the machine. The influence of commercial economy is very often not clearly understood. As is well known, the real economy of a machine is determined by the total operating expenses, among which interest on the investment and depreciation form a large, and often the largest, part. When, therefore, as an example, one machine costs $100 a year less for fuel than another, but costs $200 more in interest and depreciation, etc., evidently the first engine is not a profitable investment in spite of its high thermal efficiency.

II. Horizontal or Vertical Type

The horizontal type of engine has maintained the field since the early days, and it is usually assumed as proven that the vertical type is adapted to small sizes only. That is not true; the author is rather of the opinion that the vertical type of engine, with the crank-shaft below the cylinder, is also of great utility for the larger sizes, and is, in many important points, superior to the horizontal form. A few lines will show this. From steam-engine practice we know that vertical engines always show a somewhat better mechanical efficiency than the horizontal, and this is corroborated by the results on gas engines. Further, the losses through leaky pistons are always considerably less in the vertical than in the horizontal types. The reasons for this are obvious. The lost work due to the weight of pistons and cross-heads of horizontal machines is eliminated in the vertical type. In the latter the piston moves concentrically in the cylinder bore, with the allowable play equally distributed, and for every piston position the rings can make good contact without undue restraint. In this way the packing is made more perfect, and the piston friction, together with the amount of wear, are reduced, especially since in the vertical engine cylinder lubrication is more uniform and certain. The bending stresses occurring in the walls of the frame of horizontal machines, which can hardly be eliminated, are easily avoided in vertical engines. In the latter type the maximum pressure in the cylinder can be transmitted in an exactly central line to the crank, and the frame, thus protected against bending stresses, can be made correspondingly lighter. This point grows in importance as the piston pressures to be transmitted increase, and that explains why distinctly high-pressure engines, like the Diesel and the Bánki, adhere so persistently to the vertical form.

Besides this, the vertical type of construction gives a form of cylinder which, from the standpoints of manufacturer and of efficiency of combustion, is the most favorable, because all valves can be placed in the cylinder cover, and the combustion chamber can be kept free from all ports and dead spaces. The number of cylinders can without trouble be increased to give any desired power capacity, while at the same time a nearly complete balancing of the moving parts is possible, thus allowing of increased speeds of rotation. Since to-day internal-combustion engines are as yet mostly constructed without cross-head, the relation of the cylinder diameter D to the stroke S is not of as great importance here as it is in the case of the steam engine, and for that reason large gas engines with long stroke, or with slow speed of rotation, can be built without requiring excessive head-room. The foundations of vertical machines are affected in the main by vertical forces only. They can therefore be made smaller or cheaper than those for horizontal machines of the same power, and in some cases may even be dispensed with altogether. The taking out and replacing of pistons for cleaning purposes, etc., is easily accomplished even in vertical machines of the greatest capacity. Valves and valve gearing also are free and easily accessible, and are not, as is the case in large horizontal machines, hidden in the foundation or in trenches. Suction, exhaust, and water pipes can be put up in the most suitable manner and in the most direct way without the necessity of ditching, etc., at the place of erection. Short lengths of suction pipe, in easily accessible position, are of especial benefit in the case of suction-gas plants. Such pipes facilitate the removal of all dusty or tarry deposits, which in the case of horizontal machines, is often difficult, thus giving a better guarantee for reliability of operation. As in the cost of erection, the vertical engine is also cheaper in cost of manufacture and in maintenance and attendance. If besides this, we take into consideration that a vertical motor is free to expand axially, due to the heat generated in operation, can be most easily direct-connected to any consumer of power, and takes only about two-thirds of the floor space of a horizontal machine of the same power, we shall have touched upon all of the important advantages of this type of construction.

There are, however, cases in which the horizontal type has the preference. As long as we do not possess gas-cleaning apparatus, which is both perfect and economical, a large blast-furnace gas engine for instance should be horizontal. The reason for this is that any mechanical impurities in the gas are apt to adhere to the piston faces and cylinder bores of vertical machines, and may easily do damage. In horizontal machines, on the contrary, the combustion chamber and the valve ports may be so designed that the solid impurities of the gas can be swept out to a great extent by the exhaust gases, or can, at least, be kept away from sliding surfaces. What is said here regarding blast-furnace gas applies more or less to all industrial gases and especially to the coal-dust engine. Again, for double-acting engines the vertical is not the best form, because the placing of the valves in the lower head and their operation in this position would cause trouble. The lower head itself could be but imperfectly water-cooled. For very large power capacities the vertical type of engine is out of the question, as it is also in the case where pump or air cylinders are to be connected to the extended piston rod.

Very often the statement is made that the valve gear of vertical machines is more complex and involved than that of the horizontal. The writer does not consider this well founded, and will show later that it is possible to provide the vertical engine with a valve gear of simple and mechanically correct form. Further, it is stated that the crank-shaft of vertical engines is too much concealed, that it is hard to watch and to keep in order. To meet this it is merely necessary to point to the vertical machines of great power with which our war-ships and our merchant marine are equipped. These machines are subject to long-continued service operation under conditions much more severe than those found on land. A properly designed and well constructed crank-pin should not require anxious care if it is kept cool, protected from dust, and well lubricated. All of this, however, is accomplished with more ease in a vertical than in a horizontal machine; in the first place because the bearings are less subject to the radiant heat of the cylinder; secondly, because the frame body can be tightly enclosed; and, lastly, because the shaft so enclosed can be abundantly oiled without loss of lubricant and without the fouling due to flying grease. Even a complete enclosing of the frame does not affect the ventilation of the crank case if the interior is connected to the outer air by means of an equalization pipe. The engine room, however, will by this means be kept free from the often bothersome radiation of heat and the vapor from overheated oil.

According to all of the above, there should be little doubt that up to 150–200 I.H.P. per cylinder the vertical type of engine is, in general, fully the equal of the horizontal, and is in fact in many points superior. It is therefore to be expected that the vertical engine will have a future as a commercial gas power machine.

III. With or Without Cross-head

When, a few years after the appearance of Otto's gas engine, internal-combustion engines were no longer constructed with a cross-head, the maximum explosion pressures were from 140–170 lbs. per sq.in., while the mean pressure of the expansion rarely exceeded 50 lbs. per sq.in. At that time gas-engine practice was dominated by the wishes and requirements of the small power user, who, more strongly then than to-day, was looking for lowest prices, minimum room requirement, and greatest simplicity. In the meantime the small engine grew to be a first-class prime mover in the commercial field. The working pressures increased to from 350–500 lbs. per sq.in., the mean effective pressure to 110 lbs. per sq.in. and over. Such changed conditions naturally demanded much stricter requirements in the construction of this type of heat engine, and it would seem to the writer that among these requirements the question "With or without cross-head" deserves special consideration.

Nobody would look upon a steam engine without a cross-head as quite or entirely first class; on the contrary, stationary steam engines with trunk pistons have acquired so bad a reputation that they have one after the other disappeared from the market, or, at least, have been forced to a position of little importance. It is certainly true that one experience can perhaps not be generalized, and that these opinions of the designer of steam engines need not be considered binding by the constructor of gas engines. But a very small part of the mistrust, founded on the facts of hard experience, that the former entertains for the entire absence of cross-head, would not come amiss to the latter. The writer will try to prove this.

The omission of the cross-head in the case of small and even medium-sized gas engines, say up to 125–150 I.H.P. per cylinder, is without doubt allowable; the trunk pistons can in these cases be made so long that the unit normal pressures on the piston barrel and the cylinder wall are very small, thus affecting durability but little. Although, in this construction, cylinder lubrication requires greater attention and the piston rings are apt to blow through quicker than in case a cross-head is used, these disadvantages seldom cause trouble in commercial machines, and they are in most cases counterbalanced by the advantages of this method of construction. In the case of large engines, however, the conditions are materially different. If, in these machines, the connecting rod thrust is to be kept within limits safe for pistons and cylinder walls, and if this lateral pressure is not to interfere seriously with the work of the piston rings, the trunk piston must be made of such abnormal length as to cause trouble both in manufacture and in operation. With the size of piston we find an increase also in the internal and the expansion stresses, and in the inertia forces. The problems of properly fitting the sliding surfaces and of their lubrication finally present insurmountable difficulties. To realize this it is only necessary to remember that piston lengths exceeding 6 ft. have been proposed. How is it possible for the designer, the shop, or the operator, to absolutely control machine elements which are of such extreme size and at the same time of such vital importance? Which method of cylinder lubrication is capable of satisfactorily and economically oiling a piston surface of 60–80 sq.ft.? The union of several important operations or offices in one machine member is always a weak point, but it is entirely bad when these various offices are to each other as fire and water, i.e., when they are naturally contradictory. The most appropriate example of this practice is the trunk piston of some of our single-acting large gas engines. The purpose of the piston is in the first place to act as a machine element transmitting gas pressure without leakage, and although it is just barely able to perform this primary duty, it has besides this also been utilized as a heavily loaded machine member taking up lateral thrusts, much to the detriment of the proper performance of its chief office. That was a mistake. The piston should be designed to its smallest detail purely as an element to transmit gas pressure without leakage losses, the same should be done for the cross-head as an element to resist lateral thrust, and the two should not be united. The fact that the separate cross-head increases the length of the machine by the length of the piston rod and that the manufacturing cost is increased by a few per cent, is fully counterbalanced by better construction, less liability to repair, and greater life of the machine.

An important advantage of this construction is the single fact that the piston pin, which is usually hard to get at, difficult to keep in order, and often of barely sufficient size, is replaced by the easily accessible cross-head pin, which can be made with any desired dimensions and is easily lubricated and taken care of.

The use of a separate cross-head has no noticeable effect upon the lost work of the machine, i.e., the mechanical efficiency is not affected. This is proved by results obtained from Otto and Diesel engines. In the case of large engines even the contrary may be expected, because the coefficient of friction for a properly designed cross-head is evidently less than that for piston and cylinder surfaces highly heated, imperfectly lubricated, and subject to varying expansions and contractions. Under conditions so unfavorable to good lubrication it is really self-evident that the internal rubbing surfaces must be loaded as little as possible and consequently that they should be at the outset relieved of any lateral thrusts. The latter no longer amounts to a few hundred pounds only, in some of the largest engines it exceeds twelve and even fifteen tons during the expansion stroke, and it seems to the writer that it would pay to take care of such forces of transmission in the proper place.

As a necessary part of double-acting engines, and not on the basis of the above considerations, the cross-head again finds extended application in recent gas-engine practice. For the time being, this important step in advance is of benefit only to a few large industrial types, *but it is beyond question that in the near future all large gas-engine design will revert to the use of the separate cross-head.*

IV. Single- or Double-acting Cylinders

The first gas engines were double-acting and great consumers of gas and oil, which in a few years led to the abandonment of the use of the front end of the cylinder in order to keep the piston and cylinder bore cool by free convection. A little later this led to a considerable simplification of the entire construction in that cross-head and piston rod were also done away with. Following this, for a number of years, the double-acting principle was looked upon as impractical. For small and medium sized machines this view is probably quite justified, because to build them double-acting would make them more complex without good reason, would increase the length of the machine and raise the manufacturing cost. In such a case a two-cylinder machine is on all accounts cheaper than a double-acting single cylinder.

The case is different, however, for large machines. The complete development of this type of machine and special considerations regarding its field of application demand the greatest possible cylinder capacity, and this is best obtained economically by decreasing the piston displacement required per unit of power, that is, through a smaller number of working strokes and the use of both sides of the piston. In consequence of the more uniform crank efforts of double-acting machines, the weights of rotating parts can be made less than for single-acting engines (for the 4-cycle, for instance, the decrease is from 35–40%).

The great importance of this for large powers will be shown on page 63. A further practical advantage of double-acting machines is seen in the fact that the main driving mechanism of the engine is mechanically better utilized on account of greater specific power capacity. This, combined with the elimination of lateral thrust on the piston, causes a comparatively smaller amount of lost work, which in turn means higher fuel economy per brake horse-power for the same thermal efficiency at the cylinder.

These considerations have in the last few years led designers to again take up the construction of double-acting gas engines, and through the use of this principle several new and promising types have already been developed. The results so far obtained seem also to point strongly to a more general use of this method of operation for medium sized engines.

The employment of both sides of the piston makes it necessary, as in the case of large trunk pistons, to water-cool the piston and usually also the piston rod, and no great difficulty is encountered in doing it. The question of the stuffing-box, which was formerly so important, can to-day be considered solved for all practical purposes. As a result of experiments on the older types of single-acting tandem engines, and after years of practical experience, serviceable metallic packings for piston rods have been developed. The most difficult part of the design is found in the proper shaping of the cylinder heads, because the water-cooled stuffing-box leaves but little room for valves and igniters, and the number of openings in the head is apt to render it weak and less free to expand and contract. It is especially difficult to satisfactorily design the crank-end cylinder cover. In small engines this difficulty is not so strongly felt, because in that case the valve openings may be placed in the sides of the cylinder, thus keeping the head free and incidentally simplifying the valve gear. For large machines, however, such design is not satisfactory, for constructive and practical reasons, and should not be used, if only for the reason that the compression space is apt to be badly cut up.

In several of the more recent designs of large double-acting gas engines, however, the cylinder heads have been simplified by placing the inlet and outlet valves at the ends of the cylinder in a perpendicular plane through the cylinder axis, as is very often done in poppet-valve steam engines. It is true that by this means the cylinder heads are rendered much more satisfactory; further, it becomes easier to get at the inside of the cylinder and at the piston; on the other hand the constructive and operative difficulties before found in the heads are now transferred to the cylinder itself, i.e., to the most important part of the entire machine. Thermally, also, this valve arrangement is not satisfactory because it involves, even more so than in the case of the old form of cylinder heads, a form of combustion chamber not favorable to good combustion of the charge.[1] There is little possibility that this difficulty can ever be entirely overcome, and it is likely, therefore, that this type of large double-acting engine will labor under a certain thermodynamic disadvantage for some time to come. Practically, however, this point has little weight when, in all other respects, the double-acting type has the advantage over the single-acting.

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[1] TRANSLATOR'S NOTE. The objections to this type of cylinder seem to have been quite successfully overcome, as it is to-day used in many large engines. See Part IV.

This condition of things, i.e., complicated cylinder heads and unsatisfactory combustion chambers, can be very much relieved by employing the piston to operate the exhaust through a ring of ports at the middle of the cylinder. In such a case one valve is done away with and room is gained. Exhaust ports operated by the piston, however, can only be used in 2-cycle machines, which leads to the idea of combining the double-acting with the 2-cycle principle, as has already been done in the Körting engine. Another step in advance could be made by designing the cross-head, which in double-acting engines is a necessary part, as a pump for the air required for scavenging the cylinder, thus eliminating a separate pump. The writer first employed such a cross-head for a single-acting 2-cycle machine in the Diesel-Güldner engine. The doubling of the pump piston area required to scavenge both ends of the cylinder does not give any sensible trouble. Finally, the double-acting 2-cycle engine offers another advantage of importance in the case of large engines, in that the inertia of the reciprocating parts is taken up in the compression of the charge at the end of each stroke, and does not cause the disagreeable reversal of pressure common to all 4-cycle machines. *In a combination of the 2-cycle principle with double-acting operation we thus possess the most natural means of bringing the internal combustion-engine in an economical form to its highest capacity.*

V. Multi-cylinder Arrangements

The reason which determines the employment of several cylinders in the case of the steam engine, i.e., further decrease of fuel consumption, does not exist for the gas engine. In the case of the latter, considerations of construction and manufacture, in connection with manufacturing costs, compel a division of power between several cylinders. Let us first consider the 4-cycle engine. Although the explosion pressure occurs every fourth stroke only, and even then decreases very rapidly, it is necessary to build the driving mechanism, and in fact the entire machine, to meet the highest pressure. On that account the diameters of the shaft journals and especially the crank pin become comparatively large, which in turn means greater friction losses and which, in the case of large cylinder diameters, may lead to unwieldy and very costly constructions. This useless massing of metal, together with the increasing lost work, is the more strongly felt when we consider that the mean pressure of the 4-cycle strokes is only a comparatively small part of the explosion pressure (see p. 68). The influence of this unfavorable pressure ratio upon the driving mechanism of the engine is also strongly manifested in the weight of rotating masses required to insure the necessary uniformity of rotation. A single-acting 4-cycle engine requires a fly-wheel weight of at least 110 lbs. per brake horse-power when the peripheral speed of wheel is about 66 ft. per second and the coefficient of regulation as high as $\delta = \frac{1}{20}$. For a 500 H.P. machine of this type, therefore, the fly-wheel would weigh about 55000 lbs., which is roughly one-third of the entire engine weight. For electric lighting operation, with $\delta = \frac{1}{75}$, the engine at the same rotative speed would have to have a wheel weighing about 187000 lbs.; and for the running of alternators in parallel, with $\delta = \frac{1}{125}$, even a weight of 330000 lbs. would be required. Such enormous rotative masses increase the cost of manufacture in a manner nearly prohibitive, on the one hand, owing to their own cost, on the other to the necessary increase in the size of shafts, bearings, foundations, etc. Thus it happens that, above a certain engine capacity and a certain coefficient of regulation, two-cylinder and multi-cylinder engines can be built at less cost than single-cylinder machines of the same capacity, owing to the fact that on account of much more uniform crank efforts much lighter fly-wheels can be used.

Not less serious are the practical difficulties which oppose the increase of capacity of large single-cylinder 4-cycle engines. Parts of the machine, enormous alike in their dimensions and their weight, require in their manufacture unusually heavy machine tools and other costly shop facilities which cannot be operated economically because they are not steadily employed and are used only occasionally to their fullest capacity. For this reason, together with the fact that the manufacture of the important parts is complicated and often uncertain, the manufacturing costs become inadmissibly high and periods of delivery are greatly lengthened.

In the finished machine new difficulties are brought to light. Cylinders and cylinder covers, which, merely on account of their size and complicated form, are not easily kept free from internal stresses during the course of manufacture, are subjected to additional stresses, owing to uneven expansion under the heat developed in operation. This may easily cause a total stress exceeding the elastic limit of the material and may eventually lead to warping and rupture of the parts concerned (see pp. 120 and 131). Many thousand parts have already been lost in this way. The exact fitting of the unwieldy pistons in the cylinder bores becomes extremely difficult; too much play means constant leakage, i.e., loss of gas and of pressure, noisy operation, etc.; too tight a fit causes large friction losses through piston friction and consequent scoring of the cylinder walls. These losses are aggravated by difficulties which operate against certain and uniform lubrication of the large piston surfaces. In a limited degree water-cooling of the piston offers a useful remedy for these defects, but the gain is bought at the expense of an undesirable increase in the number of machine parts and an increase in the weight of the reciprocating masses.

Although such unfavorable mechanical conditions are in themselves sufficient to seriously affect the economy and reliability of large single-cylinder 4-cycle machines, a further heavy disadvantage comes in when we consider that the thermal operations inside of the cylinder also lose efficiency slowly as the cylinder sizes increase beyond a certain limit. Since the ratio of superficial area of the combustion chamber to its volume grows smaller as the stroke volumes increase, and the walls are made thicker from considerations of strength, less heat will be conducted through the walls and more will be stored up in them, which means that the allowable compression pressures must be steadily decreased (see p. 52). Consequently the thermal efficiency and the capacity per unit stroke volume will be correspondingly decreased, and the difficulty already encountered in satisfactorily igniting and completely burning such large charge volumes is still further magnified by the increase of compression volumes. This point is especially serious because these large machines are ordinarily run on lean gases difficult to ignite. That is the reason why in single-cylinder engines the heat supplied is thermally best utilized in sizes between 50–100 H.P., and why only in isolated cases is there a gain shown up to 150 H.P. Beyond this capacity we usually find a loss rather than a gain in economy.

Led by these considerations, German designers to-day seldom go beyond 300 B.H.P. in a single-cylinder single-acting 4-cycle machine operating on lean industrial gases. Greater capacities are obtained either by decreasing the number of strokes to a cycle or, if the 4-cycle is retained, by making it double-acting or by increasing the number of cylinders. Under such conditions the work can be handled with facility, both in the shop and the drafting room, and the machines possess a greater degree of reliability, since in case of need every cylinder, or each side of a piston, can be operated independently.

The lower limit for the division of engine capacity among several cylinders depends mainly upon the requirements of regulation; and, in the case of auto-engines, upon the required degree of balance; for the rest it is left to the judgment of the designer. In the case of stationary single-acting horizontal engines, two cylinders are usually not employed until the capacity exceeds 100–150 H.P.; for stationary vertical machines, however, the limit of division is usually lower than this. For auto-engines the deciding elements in this respect are mainly satisfactory counterbalance of the moving parts and the room available for installation.

Multi-cylinder arrangement, as applied to the construction of large units, has lost considerably in importance since the extended introduction of the double-acting type. After a short period of trial in this field these three- and four-crank constructions are considered quite as obsolete as the old so-called opposed engines, and they consequently find application to-day only in special designs, as for automobile machines, etc. In stationary practice the aim is to employ two cylinders at most, and these may be arranged either in tandem or side by side (twin engine) with the fly-wheel between them. Only the largest sizes, exceeding 1000 H.P., compel the use of four cylinders, which are usually arranged as two tandem engines side by side.

The comparative values of the usual multiple cylinder arrangements, with respect to the variation of crank efforts or the regulation, are given numerically in Tables 22 and 23, pp. 230 and 231. From these it is seen that the single-cylinder single-acting 4-cycle machine with ignitions 720° apart shows the least favorable result, while the double-acting two-cylinder 2-cycle or the double-tandem double-acting 4-cycle with ignitions every 90° of crank angle show the best. Referred to the weight of fly-wheel rim to be furnished per unit of power to two the extreme types of engine above mentioned, for otherwise exactly similar conditions, the ratio is found to be about 65 to 1. This fact natrually becomes most important in the case of large engine capacities and rigid requirements of regulation.

In general, the double-acting tandem machine has the disadvantage that the inside of the cylinders and the pistons are not easily accessible. Within certain limits this may be remedied by making the connecting piece between the two cylinders of sufficient length and of simple open form, although this means a further increase of the already great length of this type of machine and a decrease in the stiffness of the connecting member. On the other hand, the tandem machine can usually be built somewhat cheaper than a two-cylinder (side by side, so-called twin) machine of the same capacity. Another advantage of the former is that the second cylinder may be installed later, as the demand for power may require, and by doing this the coefficient of regulation for the original fly-wheel may be improved from 20 to 25% in spite of the doubling of engine capacity.

The weight of the reciprocating parts is relatively considerably greater in a tandem engine than in a twin machine, and consequently the rotative speed of the former is limited. From the standpoint of reliability of operation the tandem machine is also a little inferior to the twin, because any accident to the driving mechanism of the former, as for instance the heating of the crank pin, shuts down the entire machine, while in the case of the twin engine one cylinder may be kept in operation.

The foundations for tandem engines are always somewhat less simple than those for twin engines, but the difficulties here encountered can be overcome when the machine parts concerned have been properly built and the ground under the foundation is safe.

VI. Complete Expansion and Compounding

If the expansion volume of a charge is the same as the suction volume, that is, when the ratio of expansion is equal to the ratio of compression, which is practically always the case in engines fully loaded, the burned gases will, at the moment of exhaust, be under a pressure of from 35 to 60 lbs. which is not utilized. This loss may be diminished if the burned gases are allowed to expand beyond the suction volume, which may be done, for instance, by making the effective suction stroke shorter than the expansion stroke. This problem has occupied designers from the time of Otto, and some of the older solutions of it are found in the gas engines of Charon. The simplest solution was given by Köhler,[1] who, by closing the inlet valve before the end of the suction stroke, makes the charge volume taken in less than the piston displacement. The Körting Brothers first put this idea into practice in 1891 in their "precision" gas engines, but perfected it by putting the time of closure of the inlet valve under the control of the governor, thus suiting it to the load. Although this results in a very flexible system of speed regulation, the real aim of Köhler's idea, an increase in the thermal efficiency, was not thereby realized. The reasons for this are: At full load, with this method of regulation the suction volume is nearly or quite equal to the expansion volume; with a decreasing load the expansion becomes more and more complete, but at the same time the compression pressure decreases, and any advantage gained by the former is nearly or entirely lost by the unfavorable influence of the latter.

If complete expansion is to show economic gain, the compression pressure must remain normal for all loads; in such a case the indicated thermal efficiency is really the greater, the further such expansion is carried, p. 12. But the economic efficiency does not increase in the same ratio. Every increase in the expansion volume entails a decrease in the mean working pressure, and consequently also a lowering of engine capacity per unit volume. From this it follows that the engine mechanism is less completely utilized, and that the ratio of lost work to indicated work becomes greater. Thus, at some point, the theoretical advantage of complete expansion is changed to a practical disadvantage. How far expansion can be carried with economy cannot in general be definitely stated, but it is certain that the lower the normal compression pressure, the richer the mixture, p. 37, the slower the combustion of the charge, and the smaller the engine friction, the greater may be the ratio of expansion. Anything less than a final expansion pressure of 15 lbs. by gauge at full load is rarely used, because under such conditions the mean piston pressure or the capacity per unit volume would become too small.

In conformity with the practice of compounding in steam-engine design, attempts have from the first been repeatedly made to utilize the expansive force of the exhaust gases in a second cylinder, but always with equal failure. At best the constructed compound gas engines showed the economy of fair single-cylinder engines, but as regards simplicity, cheapness of construction, and reliability of operation, they were always inferior to the latter. The reasons for the failures of these experiments are to be found mainly in the high temperatures encountered and in the low specific heat of the burned gases. To obtain in the low-pressure cylinder a mean effective pressure at all commensurate with the friction loss, it is necessary to cut short the expansion in the first or combustion cylinder. The gases, still under high pressure and often as yet in process of combustion, then have a temperature far exceeding 1800° F., and for that reason the valves between the cylinders can be kept in fair shape only by thorough water-cooling. But this deprives the gases of a considerable proportion of their quantity of heat, and this, together with fluid friction losses, causes a decided drop in pressure. The loss of heat during the transfer of the gas is incredibly high. The writer remembers a case in which the cooling water from the two intermediate valves carried away as much heat as was transformed into indicated work, and for which the actual indicator diagrams showed only one-half the area of the ideal. It is also found that the jacket loss from the high-pressure cylinder is very large because, owing to the early exhaust opening, the mean temperature of the cylinder is considerably higher than that found in single-cylinder engines. This high mean temperature in turn has an unfavorable effect upon the lubrication of the high-pressure piston, especially since the mean pressure is also far above the normal.

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[1] Köhler, "Theorie der Gasmotoren," Leipzig, 1877, p. 22.

Taking it altogether, experiences so far hardly admit of a doubt that from the use of extended or complete expansion we can only expect an economic gain in the case of machines employing low compression, and even in such case only when the idea is carried out in one cylinder and with the simplest means. *Compounding is and will remain without promise in gas-engine construction.*

VII. Ratio of Stroke to Diameter and the Speed of Rotation

In theory purely thermal considerations appear to make it advantageous to build gas engines with shortest possible stroke and the highest possible rotative speed, in order to shorten the time of combustion and expansion to the greatest possible extent, and thus to decrease the heat losses to the cooling water. For cooling is a function of its time of action, and this is decreased when, with the same piston speed, the ratio $\frac{\text{stroke } S}{\text{diameter } D}$ is small. But for any machine, practical and not theoretical considerations have the call, and in this case practice makes demands directly opposed to theory, i.e., *greatest possible stroke and moderate speed of rotation.*

The reasons for this are various. In the first place it should be considered that a decrease in the length of the stroke means a corresponding decrease in the time available for the various operations of charging and discharging; the mixture is apt to be less uniform and the combustion less complete; the volumetric efficiency of the cylinder falls off with a consequent decrease in the charge volume. It should not be forgotten that for half of its time of operation the gas engine is an air pump and all conditions which affect the efficiency of such a pump affect the internal-combustion engine in a like manner. It is known that the volumetric efficiency of the suction stroke is, in general, but little dependent upon the rotative speed, if the stroke is long, as compared with cylinder diameter. This makes it possible for the designer to choose high speeds of rotation with high ratios of $\frac{S}{D}$, as long as he properly adapts the mixing and ignition arrangements to the high speeds. The more perfect the mixture of air and of fuel, the purer the mixture itself, and the more efficient the ignition apparatus the higher may be the piston speed—provided that thereby the efficiency of the machine as a pump is not decreased. This may be easily prevented by the use of correctly designed and properly actuated inlet and outlet valves. The largest stationary gas engines of to-day already employ piston speeds exceeding 780 ft. per minute with a ratio of $\frac{S}{D}=1.2$ to 1.3. This present limit is certainly capable of extension if the ratio $\frac{S}{D}$ is taken at a more favorable, i.e., a higher figure, and if by the expedient of making the cylinders double-acting the momentum of the reciprocating engine parts can be cushioned against the compression.[1]

The length of the stroke also has an important influence upon the shape of the combustion chamber, whose volume is of course proportional to the stroke volume. The present high compression pressures already make it difficult to so place inlet and outlet valves of sufficient size in the cylinder heads that enough internal cooling surface remains. The unfavorable effect of port openings upon the amount of cooling surface and upon the strength and elasticity of the casting itself is the more seriously felt the shorter the length of the compression space. In the case of cylinder heads of long-stroke machines, more space is available, it is easy to properly place the valves, and the port area amounts to a smaller part of the total enveloping surface of the combustion chamber.

To this must be added the fact that, for equal piston displacement, that is, for equal engine capacity, the longer the stroke the smaller will be the cylinder diameter, and consequently also the maximum distance of flame propagation. Since the length of the compression chamber is in all cases only a part of the cylinder diameter, an increase of the former at the expense of the diameter is of advantage to ignition and combustion.

The error in the use of short strokes in gas engines is, however, most strongly brought out when we examine the effect upon engine proportions. It is clear that the strength of every machine part must be computed on the basis of total piston pressure resulting from the maximum pressure of explosion. The capacity of the machine, however, does not depend upon this total maximum but upon the total mean pressure, and the metal in all parts of the machine is therefore better utilized the nearer maximum and mean pressures are together. If the mean pressure is comparatively low, the aim must be to also decrease the maximum pressures, which, with a given piston displacement or engine capacity, can only be done by decreasing the cylinder diameter and increasing the stroke. Now in the case of 4-cycle engines the ratio of mean to maximum working pressure is very low (with an explosion pressure of 350 lbs. and a mean effective pressure of 92 lbs. per sq.in., the ratio would be $350\div\frac{92}{4}=15$ to 1), and therefore the rule to obtain the necessary piston displacement with small piston area and long stroke applies most strongly to this type of engine. If this rule is neglected we arrive at machine dimensions which are uneconomical both on constructive and practical grounds. This becomes especially noticeable in the driving gear, more particularly in the crank pin, because at this point the entire force transmitted to the frame is concentrated. Diagrams, Figs. 31 and 32, show these conditions clearly. In Fig. 31 are given the crank-pin diameters for gas engines [as computed from eq. (33n), p. 166, with a maximum explosion pressure of $p_c = 350$ lbs. per sq.in.] and for steam engines (initial steam pressure 170 lbs. per. sq.in.) for various cylinder diameters. It is seen at once that the gas engine is at a considerable disadvantage. The lower curve of Fig. 32 shows the increase in crank-pin diameter for various ratios of $\frac{S}{D}$, but for equal piston displacement or equal engine capacity. Now it would of course be possible, if it is desired to use short strokes, to increase the number of revolutions instead of increasing the cylinder diameter, thus leaving the maximum total pressure the same. What that tends to, however, is indicated by the upper curve of Fig. 32, which shows the increase in peripheral velocity, v, at the surface of the crank pin for constant crank pin and cylinder diameters, and for constant piston speeds, but for decreasing ratios $\frac{S}{D}$, and consequently increasing speeds of rotation. Since the work lost in friction at the crank pin (expressed by the product $k \cdot v$) must not, for obvious reasons, exceed a certain amount, an increase in the peripheral velocity, v, calls for a corresponding decrease in the pressure, k, per unit of area. The unit pressure, however, can only be decreased by making the crank pin longer, which, in turn, calls for a larger crank-pin diameter from considerations of strength. Thus it is quite evident that an increased number of turns cannot be made to equalize the constructive disadvantages of short stroke.

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[1] The smaller automobile engines have already exceeded piston speeds of 1170 ft. per min.; as an example, the 12 H.P. double cylinder auto-engine of Remington makes 800 turns per minute, which with 4.02″ cyl. diameter and 10″ stroke, gives a piston speed of 22.3 ft. per sec. = 1338 ft. per min.

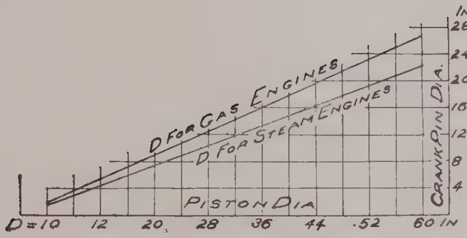

Fig. 31.—Wrist-pin Diameters for Gas and Steam Engines.

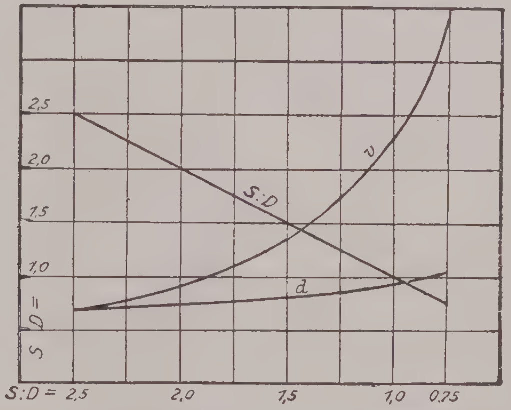

Fig. 32.—Wrist-pin Diameters and Peripheral Speeds in Gas Engines for Varying Ratios of Stroke to Diameter.

The superiority of long strokes with reference to static strength conditions is, however, by no means confined to the short period of explosion. It continues throughout the expansion period, the maximum torsional moment being smaller than in short-stroke machines of equal piston displacement in spite of the greater crank radius. Assume, for instance, that the piston displacement required for a given engine capacity is 2.65 cu.ft., and that this displacement is obtained in two engines of the following dimensions:

Case I. Cyl. diameter, 11.8″; stroke, 29.8″; ratio $\frac{S}{D} = 2.5$.

Case II. Cyl. diameter, 16.1″; stroke, 16.1″; ratio $\frac{S}{D} = 1$.

We shall then have the following results:

	Case I.	Case II.	
Max. total pressure on piston	39 000 lbs.	72 500 lbs.,	assuming 356 lbs. expl. pressure.
Pressure on piston for max. turning moment	25 300 lbs.	47 300 lbs.	at about 12% of stroke, or crank angle of 40°.
Max. tangential pressure at crank pin	20 200 lbs.	37 800 lbs.	
Max. turning moment	298 000 in.-lbs.	307 000 in.-lbs.	

This table clearly shows that in Case II the machine parts have to be made considerably heavier to withstand the greater loads, although the engines are of about the same horse-power capacity.

We thus come to the final conclusion *that the use of ratios of stroke, S, to cylinder diameter, D, as large as possible is of general advantage in internal-combustion engines*, and should be resorted to much more extensively than it is, especially when explosion pressures are high. The fact that long strokes require a somewhat greater length of machine is outweighed by the actual advantages derived, at least for stationary engines.

Petroleum engines are not excluded from this rule. It is usual to justify the use of short strokes in this type of engine by claiming that small piston travel and high speed of rotation give less chance for condensation of the kerosene vapor on the cylinder walls. In this connection one is apt to overlook the fact that in short-stroke engines the amount of kerosene drawn in is smaller (on account of lower volumetric efficiency) and that the ratio of cylinder surface to charge volume increases. Oil engines especially should therefore combine longest possible stroke with highest possible speed of rotation.

VIII. The Standard Indicator Diagram and the Allowable Stresses in Materials

It would be entirely too early, and uneconomical in construction, at this time, to adopt in general a maximum explosion pressure of from 500–560 lbs. in strength computations, although this has, on page 16, been shown to be the pressure limit up to which gain in efficiency may be expected. As yet there are only a few types of engines in which the pressure exceeds 360 lbs., while the greater majority show explosion pressures that do not greatly exceed 300 lbs. In spite of these facts, however, it is not advisable in new designs to use the common figure of 300 lbs. as a basis for computation. It is better to make p_z equal to at least 360 lbs., and even then to adopt allowable unit stresses low enough to allow explosion pressures of say 425 lbs. in case of need without endangering the construction. In this manner a free way is left open for any changes in the working diagram that may appear desirable at a later date, an early scrapping of the patterns is guarded against and their use is less restricted.

In the following strength computations the fundamental general formulæ and equations are first developed and from these, special equations (whose numbers are marked with the letter *n*) applying to the standard diagram, Fig. 33, are then derived. This standard diagram and the special formulæ based upon it simplify computations very much, allow of quick and certain insight into the principal interdependent features, and guard the inexperienced designer against faulty judgment. With a compression pressure p_c equal to 128 lbs., and a mean effective pressure of 96.5 lbs. per sq.in. (most favorable assumption) the diagram shows a maximum pressure of

356 lbs., which is taken as the basis in strength computations. With this unit pressure the maximum total pressure on the piston for a cylinder diameter of D inches will be

$$P_z = 356 \times .785\, D^2 = 279.4\, D^2 \sim 280\, D^2 \text{ lbs.}$$

With the aid of this equation the special formulæ above mentioned are derived. For the reasons already mentioned, it is not advisable to employ a lower total pressure in the computations, even if the regular use of a lower explosion pressure is intended. On the other hand, the special formulæ are no longer applicable as soon as the explosion pressure exceeds 356 lbs. The outside limit of application of these equations may be taken at 425 lbs., provided both material and workmanship are excellent. For further details of the standard diagram, see page 155.

The inertia pressures due to acceleration and retardation of the moving machine parts relieve the compression and explosion pressures on the driving gear by a few pounds per square inch of piston; but the amount of this reduction depends upon the velocity. Thus, for instance, at starting it is almost nothing, and it should not therefore be taken into consideration in strength computations. If in spite of this it should be taken into account, the working parts may be subjected to the highest stresses just at the time when the working pressures are under least control and accidental excess pressures most likely to occur.

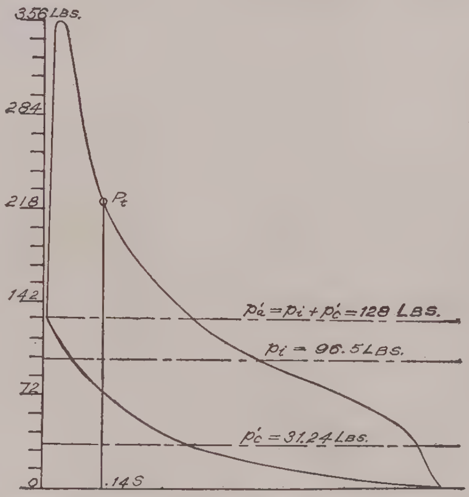

Fig. 33.—Standard Diagram used in the Derivation of the Machine Design Formulæ.

All working parts intended for engines using different kinds of fuel should be constructed for the working medium or fuel furnishing the maximum pressure, because the differences in the explosion pressures are not great enough to call for a special construction for each combustible. The gas-engine designer in any case must figure with a greater margin of safety than, for instance, the designer of steam engines. Our heat engine is not supplied, like the steam engine, with a working medium already prepared and under definite maximum pressure conditions; instead it is called upon to develop its own working pressure by means of a complex chemical process and in less than a second. It is consequently unavoidable that the assumed normal explosion pressure may in certain unforeseen instances (too early or too late ignition, formation of explosive wave due to too rich mixture, insufficient cooling, etc.) be temporarily far exceeded. Just such "supernatural forces," however, furnish the proof for conscientious design and construction of an internal-combustion engine; many a motor too economically designed has under the stress of such circumstances cost its designer very dear.

How high the allowable stresses may be chosen for the various working parts is given in each case as accurately as possible. Many examples based on successful constructions serve to show the accuracy of these assumptions. If various manufacturers use higher allowable stresses for certain materials in one or another of the machine parts, it can only be justified on the ground of superior material. The assumption of superior material cannot, however, be generalized. It is to be remembered that, in many static strength computations, stiffness, i.e., safety against elastic changes of form, is more important than mere strength. A cross-head guide, for instance, or a crank-shaft can, in spite of sufficient strength, give rise to serious interruptions of service if not designed with proper regard to stiffness or to torsional deformation. This point is especially important in the case of sliding or rotating machine parts and for surfaces supporting any important packings.

Be on the safe side! Far-fetched refinements have no place in strength computations; because the estimation of the cause and of the magnitude of the stresses in constructions not yet built depends generally upon assumptions which in reality are never quite true. At the outset even the properties of our principal materials of construction are subject to uncertain variation. Internal stresses and changes induced in them during the process of manufacture and afterward escape definite consideration altogether. Considering this and the occasional unavoidable accidental occurrences in operation, it will be clear that in our strength computations the factor of safety should be considered of more importance than the factor of "accuracy," and that, for the rest, the main aim should be to make the work as clear and to the point as possible.

B. DETERMINATION OF THE PRINCIPAL DIMENSIONS

I. Cylinder Diameter D and Stroke S on the Basis of Thermodynamic Laws

It is apparently but a step from the fundamental thermodynamic laws discussed in Part II to the determination of the cylinder dimensions required for any given engine capacity, but just here is where our present thermodynamic knowledge fails us. It is true, of course, that the theoretical pressure diagram of the gas engine is almost definitely defined, and the area developed so logically fixed that it is quite easy to determine the amount of work done either by means of it or by the aid of fundamental equations. But this does not serve the designer, since he has to deal not with the theoretical but with the actual indicated diagram. The actual diagram, however, is subject to so long a list of accidental variations (as gas content, purity, and temperature of mixture, state of mixing and kind of combustion, location and effect of ignition, form of combustion chamber, amount of the heat losses, etc.) that it is next to impossible to draw even approximate comparisons in this case between theory and practice. If this is attempted we are apt to end up with a number of scattering factors of estimation and judgment which make it easy to compute any desired result except the correct one, and to which the man at the drawing board becomes accustomed only with difficulty, because the deductions are mostly based on the somewhat remote ground of strict thermodynamic theory.

The ratio of the actually indicated work to that shown by the theoretical diagram, which in conformity with steam-engine practice we will call the *card factor*, varies between .4 and .8, i.e., the variation is not less than twice the lower limiting value. Under such conditions no reliable estimation of the true value is possible. If therefore it is desired to obtain easy and practically useful design formulæ for the determination of the principal dimensions, their derivation should not be based on the theoretical card or upon the general fundamental equations for gases. The designer is then forced to use a practical method of computation in which the accuracy of the final result depends, as usual, first upon the judgment of the estimator, and secondly, upon the correct inter-relation of the various factors of experience that enter the problem.

II. Dimensions D and S according to the Quantity of Air required for Combustion

The simplest and most certain basis for the determination of the cylinder dimensions required for any given engine capacity is found in the amount of air necessary for combustion. *Air is power* for the engine builder, for the two are in direct proportion. (Doubling the weight of air drawn in allows of a doubling of the weight of fuel, increasing the engine capacity in the same ratio). Now the amount of air per stroke available for combustion depends only upon the details of construction, and since these are under the control of the designer in every case, the method of computing the possible capacity of the engine from the amount of air used in unit time, or, inversely, of computing the cylinder dimensions required to generate the volume equal to the certain volume of air necessary for any given capacity, is to him the most natural. This method of computation allows of the fullest consideration of the properties of all the various fuels; it requires but very few assumptions, and those that must be made are easy of clear determination in the light of experience.

In the following, a method of computation for the determination of the cylinder dimensions based upon the considerations above stated, is briefly outlined. Its practical applicability to gas engines of all types and sizes has been demonstrated for some years. The scheme incidentally has another advantage in that two values (L_h and C_h), which are used later on, for the determination of valve and pipe dimensions, are determined at the outset, once for all.

1. General Fundamental Equations.

Let N_n = nominal brake horse-power;

n = r.p.m.;

D = piston diameter in feet;

S = stroke of piston in feet;

$V_h = .785D^2S$ = piston displacement in cubic feet;

$V = \eta_e V_h$ = the volume of mixture under standard conditions actually drawn in (29.9″ Hg. and 32° F.);

L = the practically most favorable amount of air required per cubic foot of gaseous or per pound of liquid fuel, in cubic feet. This is not the theoretical amount of air required for the fuel, but the theoretical plus such excess as seems best in practice;

L_h = the actual amount of air required in cubic feet per suction stroke of the engine, determined from L and for the normal power N_n;

C_s = the amount of fuel used per hour at the normal load N_n, for gases in cubic feet, for liquids in pounds;

C = the amount of fuel per H.P. hour (otherwise same conditions as for C_s);

C_h = the amount of fuel per suction stroke (other conditions as under C_s);

H = the lower heating value of the fuel in B.T.U., for gases per cubic foot, for liquids per pound;

$\eta_e = \frac{V}{V_h}$ = volumetric efficiency of the suction stroke (referred to standard conditions);

η_w = economic efficiency = thermal efficiency at the brake $= \frac{N_n \times 33\,000 \times 60}{778 C_s H} = \frac{2545 N_n}{C_s H}$.

FOR FOUR-CYCLE ENGINES (per cylinder end):

$$C_s = \frac{N_n \times 33\,000 \times 60}{778 H \eta_w} = \frac{2545 N_n}{H \eta_w}; \qquad (1)$$

$$C_h = \frac{2C_s}{60n} = \frac{2 \times 2545 N_n}{60 n H \eta_w} = \frac{84.8 N_n}{n H \eta_w}; \qquad (2)$$

$$L_h = \frac{C_s L}{30n} = \frac{2545 N_n L}{30 n H \eta_w} = \frac{84.8 N_n L}{n H \eta_w}. \qquad (3)$$

FOR TWO-CYCLE ENGINES equations (2) and (3) should be divided by 2, because there is a charging stroke every revolution.

(*a*) **Engines for Gas Fuel.** The actual charge drawn in during one suction stroke must be

$$V = C_h + L_h.$$

This volume requires an actual piston displacement of

$$V_h = .785 D^2 S = \frac{C_h + L_h}{\eta_e} = \frac{84.8 N_n + 84.8 N_n L}{n H \eta_e \eta_w}$$

$$= \frac{84.8 N_n (1+L)}{n H \eta_w \eta_e} \text{ cubic feet.} \qquad (4)$$

Solving (4) for D, S, and n in turn, we have

$$D = \sqrt{\frac{108 N_n (1+L)}{n H S \eta_w \eta_e}} \text{ feet;} \qquad (5)$$

$$S = \frac{108 N_n (1+L)}{n H D^2 \eta_w \eta_e} \text{ feet;} \qquad (6)$$

$$n = \frac{108 N_n (1+L)}{H D^2 S \eta_w \eta_c} \text{ r.p.m.} \qquad (7)$$

(*b*) **Engines for Liquid Fuel.** In these engines the fuel is introduced into the cylinder either as liquid or vapor. But even in the state of vapor the ratio of fuel volume to air volume in the mixture is much smaller than it is with the richest gases. For example, in the case of illuminating gas, about 5.5 volumes of air per volume of gas are theoretically required; for benzene vapor the ratio is 45 to 1. Even crude alcohol, comparatively low in heating value, theoretically occupies only about 4% of the volume of the mixture, and on account of the excess of air the real figure is between 2 and 3%. In view of this fact the volume occupied by the vapor in the fuel mixture may be neglected, and we may write directly

$$V_h = .785D^2S = \frac{L_h}{\eta_e} = \frac{84.8N_nL}{nH\eta_w\eta_e}. \quad \ldots \ldots \ldots \quad (8)$$

Solving (8) for D, S, and n, we obtain for engines using liquid fuel,

$$D = \sqrt{\frac{108N_nL}{nHS\eta_w\eta_e}} \text{ feet}; \quad \ldots \ldots \ldots \quad (9)$$

$$S = \frac{108N_nL}{nHD^2\eta_w\eta_e} \text{ feet}; \quad \ldots \ldots \ldots \quad (10)$$

$$n = \frac{108N_nL}{HD^2S\eta_w\eta_e} \text{ r.p.m.} \quad \ldots \ldots \ldots \quad (11)$$

Equations (4) to (11) do not contain a single factor that may not in every case be determined with accuracy. Depending completely on judgment only are the factors η_w and η_e, and in view of the wealth of experience available, a mistake with reference to these two is hardly possible. Every designer knows what thermal efficiency on the brake (η_w) he may demand for the given size of engine and the given kind of fuel. A moment's consideration will also show him that, for instance, for a high-speed machine with automatic inlet valve and imperfect cooling, the volumetric efficiency (η_e) must be chosen lower than for good engines of larger size. Values of η_e based upon experience will be found on p. 31, Part I. For η_w, H, and L, Table 7, p. 76, gives average practical figures. In this table the figures in columns 3 to 8 refer to the nominal H.P. (N_n) of the engine. Since this capacity N_n in good constructions is from about 15 to 20% less than possible maximum capacity, and since for this $N_{\max}$ a sufficient excess of air is required, the amount of air required for N_n should never be taken into computation at less than 30% excess over the theoretical amount. But high-compression pressures, especially in connection with the rich liquid fuels, require a still further dilution of the mixture, and amounts of air in excess from 50 to 60% are common. For obvious reasons the amount of air used can be nearer the theoretical the more perfect the mixture, the lower the compression, and the more effective the cooling. A perfectly uniform mixture is difficult to obtain in the case of liquid fuel engines; the fuels used by them are very rich throughout, and, on account of the usually very complex charging actions, they are much more sensitive against overload than other machines. All of this compels the use of a greater excess of air in liquid fuel engines than in others, and this has been taken into account in the figures given in the table.

The fuel consumption, C (columns 4 to 8) presupposes up-to-date construction and proper practical conditions of operation; on the testing floor and in acceptance tests better figures are very often obtained. Of course there are still several types which appear to be satisfied with lower economy. The consumption of ignition and heating apparatus has no connection with the internal operation, and has consequently not been taken into account in determining C. In the case of power-gas engines it is assumed that suction generators are used, in which case separate steam-boilers are not required and the generator itself furnishes the heat required for the vaporization of the water. The consumption of the gaseous fuels is based on 28.9″ Hg. pressure and 59° F.

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TABLE 7

Column No.	1		2		3		4			5			6			7			8		
	Lower Heating Value (H) B.T.U. per		Air Required, Cubic Feet.				Consumption of Fuel (C) per Brake Horse-power Hour, and Thermal Brake Efficiency (ηw), when N_n=														
			Theoretically. (L_0) per		Actually. (L) per		5 B.H.P.			10 B.H.P.			25 B.H.P.			50 B.H.P.			100 B.H.P. and over		
							(C)		(ηw)	(C)		(ηw)	(C)		(ηw)	(C)		(ηw)	(C)		(ηw)
	Cu.Ft.	Lb.	Cu.Ft.	Lb.	Cu.Ft.	Lb.	Cu.Ft.	Lbs.		Cu.Ft.	Lbs.		Cu.Ft.	Lbs.		Cu.Ft.	Lbs.		Cu.Ft.	Lbs.	
I. Illuminating gas.																					
a. Lean	505		5.5		7.5		24.8		.20	22.2		.22	20.4		.24	19.1		.26	18.5		.27
b. Ordinary	560		to		to		22.2		.20	20.1		.22	18.3		.24	16.9		.26	16.5		.27
	618						20.4		.20	18.3		.22	16.9		.24	15.5		.26	15.1		.27
c. Rich	675		6.5		10.0		18.7		.20	16.7		.22	15.5		.24	14.1		.26	13.7		.27
II. Generator gas.																					
a. On basis of anthracite[1]		13 400						1.65	.11		1.43	.13		1.25	.15		1.10	.17		1.03	.18
b. Anthracite gas	141		.85		1.1		106.0		.17	95.3		.19	84.8		.21	77.6		.23	73.2		.24
c. On basis of coke[1]		12 950	to		to			1.76	.11		1.49	.13		1.29	.15		1.14	.17		1.07	.18
d. Coke gas	129		1.00		1.4		116.0		.17	102.0		.19	91.8		.21	84.8		.23	81.3		.24
III. Blast-furnace gas	106		.75		1.0–1.2					130.0		.18	116.0		.20	102.0		.22	98.8		.24
IV. Coke-oven gas	505		5.3		7.0					35.3		.17	30.0		.19	26.4		.21	24.7		.23
V. Kerosene		18 900		185.0		257–353		1.25	.11		1.10	.12		.99	.13						
VI. Crude oil (Diesel)		18 000		176		288–323		.55	.25		.527	.26		.506	.27		.462	.30		.440	.315
VII. Gasolene		18 500		176		240–323		.66	.19		.615	.21		.550	.23						
VIII. Alcohol (90% vol.)		10 300		96.5		128–193		1.10	.22		.99	.24		.92	.26						

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[1] Including 10–15% of the full daily consumption for firing-up. The consumption of a generator banked through the night is from one-half to one-third of this amount.

2. Special Equations for the Principal Fuels. From the fundamental equations (4) to (11) we may derive useful special equations for the main gas-engine fuels by substituting in them average values for the heating value (H) and for the air required (L). Only the factors η_w and η_e then remain unknown, and these values for the given conditions may be taken from the tables. In the derivation of the following special equations (12) to (29), which apply to slow-speed stationary engines of fair quality, one step further has been taken in that η_e has been assumed $=.85$ for this type of engine. For *this* kind of machine, therefore, the designer is called upon merely to estimate the value of η_w for the size of the engine and to determine the fuel to be employed. For types differing from the above, high-speed automobile engines, for instance, the fundamental equations (4) to (11) should be used, on account of the smaller values for η_w and η_e.

(*a*) ENGINES FOR GAS FUEL.—*Illuminating Gas* (from bituminous coal):

$H=565$ B.T.U. per cu.ft.; $L=8.5$ cu.ft. per cu.ft.

$$D=\sqrt{\frac{2.14N_n}{Sn\eta_w}}\text{ feet;} \qquad (12)$$

$$S=\frac{2.14N_n}{D^2n\eta_w}\text{ feet;} \qquad (13)$$

$$n=\frac{2.14N_n}{D^2S\eta_w}\text{ r.p.m.;} \qquad (14)$$

Power Gas (Producer Gas):

$H=135$ B.T.U. per cu.ft.; $L=1.30$ cu.ft. per cu.ft.

$$D=\sqrt{\frac{2.16N_n}{Sn\eta_w}}\text{ feet;} \qquad (15)$$

$$S=\frac{2.16N_n}{D^2n\eta_w}\text{ feet;} \qquad (16)$$

$$n=\frac{2.16N_n}{D^2S\eta_w}\text{ r.p.m.} \qquad (17)$$

Blast-furnace Gas:

$H=108$ B.T.U. per cu.ft.; $L=1.1$ cu.ft. per cu.ft.

$$D=\sqrt{\frac{2.47N_n}{Sn\eta_w}}\text{ feet;} \qquad (18)$$

$$S=\frac{2.47N_n}{D^2n\eta_w}\text{ feet;} \qquad (19)$$

$$n=\frac{2.47N_n}{D^2S\eta_w}\text{ r.p.m.} \qquad (20)$$

Examples. 1. Illuminating-gas engine, Otto-Crossley.[1] $N_n=35$ B.H.P., $D=13.2''$ (1.1 ft.), $S=20.9''$ (1.74 ft.), $n\sim163$ r.p.m., $H=547$ B.T.U. per cu.ft., $\eta_w=.208$. Assume $L=8.5$ cu.ft., and $\eta_e=.85$ Then according to equation (5),

$$D=\sqrt{\frac{108\times35\times9.5}{163\times547\times1.74\times.208\times.85}}=1.14 \text{ ft.}=13.7''.$$

According to special eq. (12),

$$D=\sqrt{\frac{2.14\times35}{1.74\times163\times.208}}=1.126 \text{ ft.}=13.5''.$$

2. Old Type Producer-gas Engine, Otto-Deutz.[2] $N_n=150$ B.H.P. in two cylinders, $D=20.5''$ (1.71 ft.), $S=30''$ (2.5 ft.), $n\sim140$ r.p.m., $\eta_w=.115$ on the basis of coke=about .16 referred to producer-gas. $H\sim130$ B.T.U. per cu.ft., assume $L=1.25$ cu.ft. per cu.ft., and $\eta_e=.85$.

According to eq. (5),

$$D=\sqrt{\frac{108\times75\times2.25}{140\times130\times2.5\times.16\times.85}}=1.72 \text{ ft.}=20.6''.$$

According to special equation (15),

$$D=\sqrt{\frac{2.16\times75}{2.5\times140\times.16}}=1.71 \text{ ft.}=20.5''.$$

3. Modern Suction Gas-engine, Körting Bros., $N_n=100$ B.H.P., $D=20.5''$ (1.71 ft.), $S=27.6''$ (2.3 ft.), $n=140$ r.p.m.

With the general assumptions made, we will have by special eq. (15),

$$D=\sqrt{\frac{2.16\times100}{2.3\times140\times.24}}=1.68 \text{ ft.}=20.2''.$$

4. Suction-gas Engine, Güldner. $N_n=100$ B.H.P., $D=18.7''$ (1.56 ft.), $S=27.6''$ (2.3 ft.) $n=160$ r.p.m.

From special eq. (15),

$$D=\sqrt{\frac{2.16\times100}{2.3\times160\times.24}}=1.53 \text{ ft.}=18.4''.$$

5. Blast-furnace Gas Engine, Körting Bros.[3] $N_n=100$ B.H.P., $D=19.7''$(1.64 ft.), $S=34.5''$ (2.87 ft.), $n=130$ r.p.m., $H=112$ B.T.U. per cu.ft., $C=91.7$ cu.ft. per B.H.P. from which $\eta_w=.24$. With $\eta_e=.85$ and $L=1.1$ cu.ft. per cu.ft., we will have, according to eq. (5),

$$D=\sqrt{\frac{108\times100\times2.1}{130\times112\times2.87\times.24\times.85}}=1.63 \text{ ft.}=19.6''.$$

From special eq. (18),

$$D=\sqrt{\frac{2.47\times100}{2.87\times130\times.24}}=1.66 \text{ ft.}=19.9''.$$

Since eq. (18) assumes $H=108$ B.T.U. in its derivation, the value for D is in this case naturally somewhat greater than by eq. (5).

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[1] Schöttler, Die Gasmaschine, 3d ed., p. 167.

[2] Schöttler, Die Gasmaschine, 3d ed., p. 172.

[3] Stahl und Eisen, 1900, p. 413.

6. Blast-furnace Gas Engine, Cockerill.[1] $N_n=600$ B.H.P., $D=51.2''$ (4.26 ft.), $S=55.0''$ (4.59 ft.), $n=95$ r.p.m., $H=108$ B.T.U. per cu.ft., $\eta_w=.19$, η_e and L assumed as before. Then from (5),

$$D=\sqrt{\frac{108\times600\times2.1}{95\times108\times4.59\times.19\times.85}}=4.24\text{ ft.}=51.0''.$$

According to special eq. (18),

$$D=\sqrt{\frac{2.47\times600}{4.59\times95\times.19}}=4.25\text{ ft.}=51.1''.$$

(*b*) ENGINES FOR LIQUID FUEL.—*Kerosene:*

$$H=18\,900\text{ B.T.U. per lb.};\quad L=300\text{ cu.ft. per lb.}$$

$$D=\sqrt{\frac{2.02N_n}{nS\eta_w}}\text{ feet;} \qquad (21)$$

$$S=\frac{2.02N_n}{nD^2\eta_w}\text{ feet;} \qquad (22)$$

$$n=\frac{2.02N_n}{D^2S\eta_w}\text{ r.p.m.} \qquad (23)$$

Gasoline:

$$H=19\,000\text{ B.T.U. per lb.};\quad L=280\text{ cu.ft. per lb.}$$

$$D=\sqrt{\frac{1.87N_n}{nS\eta_w}}\text{ feet;} \qquad (24)$$

$$S=\frac{1.87\,N_n}{nD^2\eta_w}\text{ feet;} \qquad (25)$$

$$n=\frac{1.87\,N_n}{D^2S\eta_w}\text{ r.p.m.} \qquad (26)$$

Crude Alcohol, about 90% by volume:

$$H=10\,250\text{ B.T.U. per lb.};\quad L=160\text{ cu.ft. per lb.}$$

$$D=\sqrt{\frac{1.98N_n}{nS\eta_w}}\text{ feet;} \qquad (27)$$

$$S=\frac{1.98N_n}{nD^2\eta_w}\text{ feet;} \qquad (28)$$

$$n=\frac{1.98N_n}{D^2S\eta_w}\text{ r.p.m.} \qquad (29)$$

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[1] Stahl und Eisen, 1900, p. 721.

Examples. 1. Kerosene Engine, Grob & Co.[1] $N_n=8$ B.H.P., $D=9.05''$ (.755 ft.), $S=9.05''$ (.755 ft.), $n=266$ av., $H=19\,850$ B.T.U. per lb., $\eta_w=.136$. Assume $L=300$ cu.ft. per lb., and on account of automatic inlet valve $\eta_e=.83$. Then according to eq. (9),

$$D=\sqrt{\frac{108\times 8\times 300}{266\times 19\,850\times .755\times .136\times .83}}=.76\text{ ft.}=9.0''.$$

and from special eq. (21),

$$D=\sqrt{\frac{2.02\times 8}{266\times .755\times .136}}=.77\text{ ft.}=9.25''.$$

2. Alcohol Engine, Altman & Co.[2] $D=11.0''$ (.92 ft.), $S=15.8''$ (1.31 ft.), $n=200''$. With alcohol alone N_n was 18 B.H.P. With a benzol-alcohol mixture η_w varied between .13 and .17. Assuming the same efficiency of heat transfer gives an average $\eta_w=.15$ for alcohol alone. Then, with $L=160$ cu.ft. per lb., eq. (9) gives

$$D=\sqrt{\frac{108\times 18\times 160}{200\times 10\,250\times 1.31\times .15\times .85}}=.954\text{ ft.}=11.45''.$$

For special eq. (27),

$$D=\sqrt{\frac{1.98\times 18}{200\times 1.31\times .15}}=.95\text{ ft.}=11.40''.$$

The close agreement of the computed with the actual dimensions shows the general applicability of the method above developed. In order to extend it to 2-cycle machines it is only necessary to introduce into the formula a factor representing the ratio of capacities. This type of machine works throughout with special charging pumps whose discharge capacity of course directly affects engine capacity. With pumps of sufficient size a means is given of considerably increasing the weight of air per suction stroke for the combustion process by merely raising the transfer pressure. The effective power of the engine, however, does not increase in the same ratio because the increase in the air supply entails a higher exhaust pressure and increased work in the pumps, which means a decrease in the mechanical efficiency of the engine. Both of these effects can be taken into account by suitably changing the value of η_w.

The special equations give an insight into the *ideal* ratios of specific engine capacity for the various kinds of engines, assuming η_w the same for all types. To judge of the *real* ratio of capacity, the actual value of η_w must be considered. Thus, for example, the ideal capacity ratio between an illuminating-gas and a producer-gas engine, assuming η_w equal, would be as 2.16:2.14, from eqs. (12) and (15). The difference is very small. In practice, however, the former may be expected to give a value of η_w of about .27, while the latter only gives .24, so that in reality the producer-gas engine would have to be larger than the illuminating-gas engine by about $\frac{.27-.24}{.27}=10$ to 12%. This does not mean that it is not possible to obtain the same B.H.P. from any given illuminating-gas engine when operated with producer-gas, if only the engine is of ample size for the power and the combustion is not too imperfect, so that the values of η_w are not too far apart. It is a fact, however, that these conditions are found only in a few excellently designed and well constructed machines.

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[1] Z. d. V. D. I., 1895, p. 616.

[2] Ihering, Die Gasmaschine, 2 ed., p. 301.

III. Dimensions D and S according to Practical Capacity Coefficients

The following practical capacity coefficients are all based on experience with machines actually constructed. Their use therefore presupposes similar conditions. For special cases, or for those which require strict treatment, these empirical coefficients can be relied upon to only a small extent. In such cases it is better to use the method outlined under II.

Table 8 gives the most common capacity coefficients. The figures refer to the nominal brake horse-power, N_n, of 4-cycle machines. For maximum load, therefore, they should be increased in corresponding proportion. For 2-cycle engines the specific capacity L_0 of 4-cycle machines (that is, the work done per second per cu.ft. of displacement) may be multiplied by from 1.75 to 1.90.[1] The specific piston displacement V_0 (that is, the cubic feet piston displacement required per second to develop one horse-power) is thereby decreased in the inverse ratio.

TABLE 8

CAPACITY COEFFICIENTS FOR 4-CYCLE ENGINES

Assume Mechanical Efficiency $\eta_m=.80$.	M.E.P. p_i. Pounds per Square Inch.	$p_n=p_i\eta_m$. Pounds per Square Inch.	$L_0=\frac{144p_n}{4}$. Foot-pounds.	$V_0=\frac{550}{L_0}$. Cubic Feet.	$K^*=.0017p_n$	Relative Capacity Referred to Illuminating-gas Engine =1.00.
Illuminating-gas engine	78.0	62.5	2250	.245	.106	1.00
Producer-gas engine	60.5	48.4	1740	.316	.082	.77
Blast-furnace gas engine	57.0	45.6	1640	.335	.078	.73
Gasolene engine	74.5	59.6	2150	.256	.102	.95
Kerosene engine	54.5	43.6	1570	.350	.074	.70
Diesel oil engine	99.5	79.6	2870	.192	.136	1.27
Alcohol engine	54.5	43.6	1570	.350	.074	.70

* See page 83.

By substituting the above capacity coefficients in the formulæ following (eqs. 30 to 37), the principal dimensions may be determined. The notation used in the derivation of these formulæ is the same as that given on p. 73.

1. Indicated mean effective pressure p_i in lbs. per sq.in.

$$\text{Nominal B.H.P., } N_n=\frac{144p_i\eta_m\times .785D^2Sn}{33\,000\cdot 2}$$

$$=\frac{p_i\eta_m D^2Sn}{585}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (30)$$

---

[1] TRANSLATOR'S NOTE. It is evident that this statement can apply to medium and large sized 2-cycle machines with separate pumps only. It is a well known fact that the power of small high-speed 2-cycle engines with crank case compression is often no greater than that of 4-cycle machines of equal cylinder diameter. This is due mainly to low volumetric efficiency of the small 2-cycle cylinder.

Or, if we let F = area of piston in sq.ft., and $c = \frac{Sn}{30}$ = piston speed in feet per second, we may also write

$$\text{Nominal B.H.P., } N_{\iota} = \frac{144 p_i \eta_m F c}{550 \times 4} = .066 p_i \eta_m F c = .066 p_i \eta_m V_s, \quad . \quad . \quad . \quad (30a)$$

in which $V_s = Fc = \frac{.785 D^2 S n}{30}$ cu.ft. = piston displacement per second.

From eq. (9), p. 8. we know that the piston displacement V_h required per stroke for N_i indicated horse-power at n r.p.m. is

$$V_h = \frac{33\,000 N_i}{144 p_i \frac{n}{2}} = \frac{458 N_i}{p_i n} \text{ cu.ft.}$$

For *one horse-power*, therefore, the required displacement per stroke is

$$V_h = \frac{458}{p_i n} \text{ cu.ft.}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (31)$$

and for any given cylinder dimensions the indicated horse-power is

$$N_i = \frac{p_i n}{458} V_h. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (31a)$$

It is evident that the factor $\frac{p_i n}{458}$ expresses the I.H.P. generated by one cubic foot of piston displacement in 4-cycle engines, and conversely $\frac{458}{p_i n}$ measures the cubic feet of piston displacement required for every I.H.P. These two reciprocal factors become equal to unity, that is, one cubic foot of piston displacement will generate one I.H.P., when $p_i n = 458$. In the average machine with p_i = say 75 lbs. per sq.in. and $n \sim 180$ r.p.m., $p_i n$ will equal about 13 500, which means that each cubic foot of piston displacement in a 4-cycle machine will furnish about 29 I.H.P.[1]

If the engine operates on the hit-and-miss system of regulation so that there are z explosions per minute, the I.H.P. of any partial load will be

$$N_i = \frac{144 p_i \times .785 D^2 S z}{33\,000} \sim .0034 p_i D^2 S z. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (32)$$

Assuming that the indicated mean effective pressure of the suction and exhaust strokes combined is p_w lbs per sq.in., the work of charging and discharging the cylinder will be

$$N_w = \frac{144 p_w \times .785 D^2 S \left(\frac{n}{2} - z\right)}{33\,000} = .0034 p_w D^2 S \left(\frac{n}{2} - z\right). \quad . \quad . \quad . \quad . \quad . \quad . \quad (33)$$

---

[1] For single cylinder steam engines eq. (31) has the form $V_h = \frac{113}{p_i n}$. For the average case $p_i = 30$ to 40 lbs. per sq.in. When it is further considered that the economical number of turns for the steam engine is about one-third less that for the gas engine, the ratio of the specific piston displacement between gas and steam power will be about as 1.0 to .75, which is not as unfavorable to the gas engine as is generally assumed.

Consequently the net I.H.P. of the engine will be

$$N_i'=N_i-N_w=.0034D^2S\left[zp_i-\left(\frac{n}{2}-z\right)p_w\right]. \qquad (32a)$$

The indicated I.H.P. gives a higher mechanical and a lower thermal efficiency than the gross I.H.P., since $N_i>N_i'$. In regard to this see page 8,

2. Useful mean effective pressure p_n lbs. per sq.in. for N_n B.H.P.

$$\text{Nominal B.H.P., } N_n=\frac{144p_n\cdot .785D^2Sn}{33\,000\cdot 2}=\frac{p_nD^2Sn}{585}, \qquad (34)$$

or

$$N_n=\frac{144p_nFc}{550\times 4}=.066p_nFc=.066p_nV_s. \qquad (34a)$$

Conversely, therefore, $p_n=\dfrac{N_n}{.066Fc}=\dfrac{N_n}{.066V_s}$ lbs. per sq.in.

3. Specific capacity, L_o, in ft.-lbs. per cu.ft. of piston displacement per second.

$$\text{Nominal B.H.P., } N_n=\frac{L_0\cdot .785D^2Sn}{550\times 30}=.000048L_0D^2Sn=\frac{L_0Fc}{550}. \qquad (35)$$

4. Specific piston displacement, V_o, in cu.ft. per second for 1 B.H.P.

$$\text{Nominal B.H.P., } N_n=\frac{.785D^2Sn}{30V_0}=\frac{Fc}{V_0}=\frac{D^2Sn}{38.5V_0}, \qquad (36)$$

from which

$$V_0=\frac{Fc}{N_n}=\frac{D^2Sn}{38.5N_n}.$$

5. Capacity constant K:

$$N_n=KD^2Sn \text{ B.H.P.} \qquad (37)$$

in which the constant $K=\dfrac{144p_n\times .785}{33\,000\times 2}=.0017p_n$ from the general equation

$$N_n=\frac{144p_n\times .785D^2Sn}{33\,000\cdot 2}.$$

Eq. (37), on account of its great simplicity, is especially well adapted for rapid computation or for comparison of capacities.

A useful aid for the determination of D and S is given in the diagram, Fig. 34. The curves are constructed for a mean $p_n=65$, or $p_i=$about 77.5 lbs. per sq.in. and for 200 r.p.m. The best way to use this diagram is to determine first not the nominal but the maximum capacity which the engine is to have (since 77.5 lbs. per sq.in. is near the upper limit for average construction) and then to find proper values of D and ρ $\left(\rho=\dfrac{S}{D}\right)$ from the curves. If instead of 200 r.p.m., the figure assumed in the diagram, any other number of turns is desired, the maximum capacity must be adjusted to this figure.

Example. A 100 B.H.P. engine is to have an excess capacity of 20%, at 150 r.p.m. This means a maximum B.H.P. of 120, but this must be changed to $120\dfrac{200}{150}=160$ B.H.P. to get it to a 200 r.p.m. basis. Assuming $\rho=\dfrac{S}{D}=1.6$, the upper set of curves under B.H.P.$=160$ gives $D=19.9''=$say $20''$. The data for the 100 B.H.P. engine will then be $D=20''$, $\rho=1.6$ $S=1.6\times 20=32''$, r.p.m.$=150$.

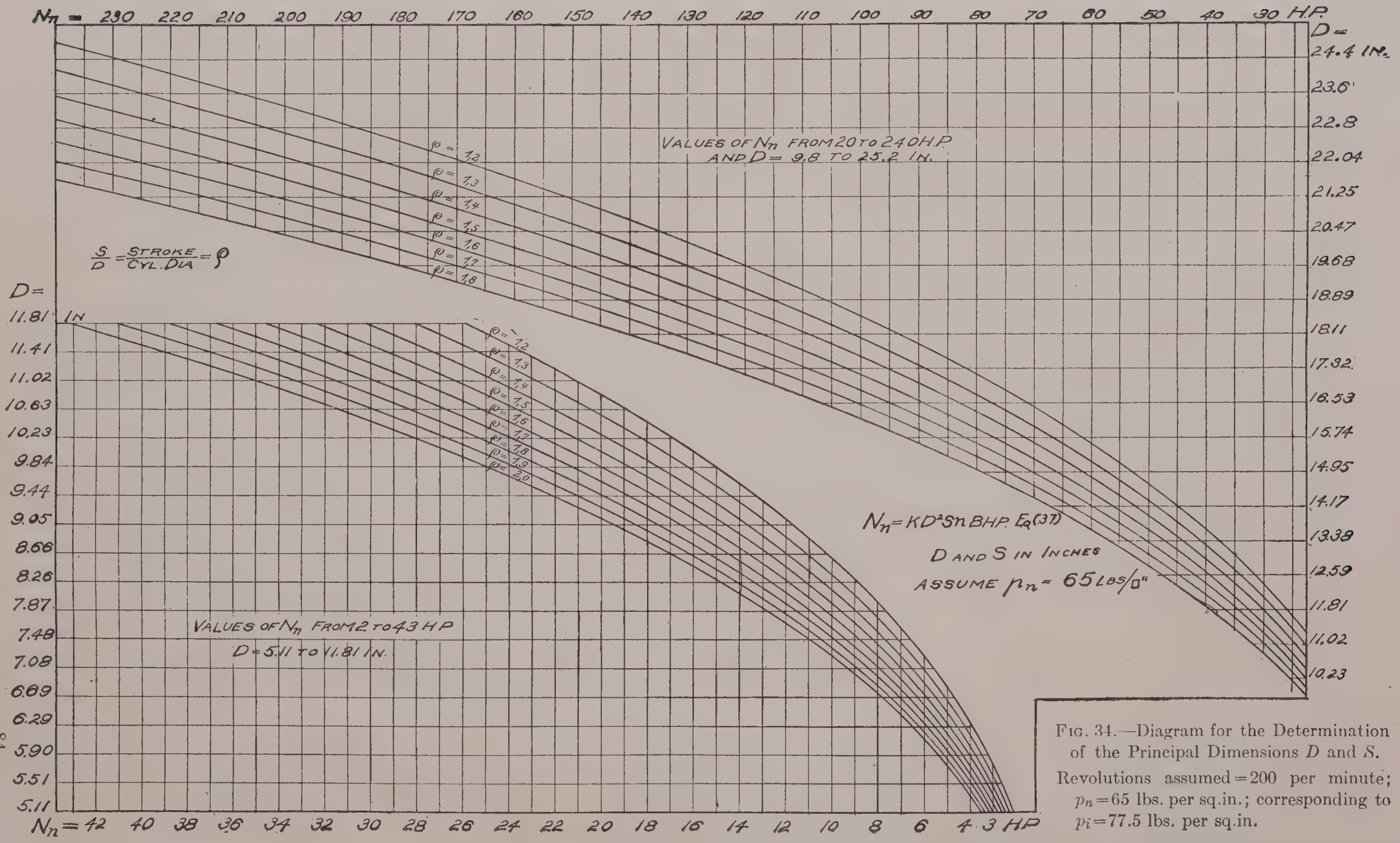


FIG. 34.—Diagram for the Determination of the Principal Dimensions D and S. Revolutions assumed = 200 per minute; p_n = 65 lbs. per sq.in.; corresponding to p_i = 77.5 lbs. per sq.in.

C. GENERAL ENGINE PARTS

By general parts are meant those used for gas as well as for oil engines; according to the kind of fuel employed these are supplemented by special parts, which are taken up in the next chapter. Finally, every engine installation requires certain auxiliaries which, although they are not a part of the machine proper, are generally built and furnished by the engine manufacturer, and are for that reason considered in Part E, p. 268.

The designs shown in the following construction plates are for the main part taken from modern and tried types of engine, but they should not on that account be taken as models without exception. Special circumstances sometimes require and sanction constructive means, which are inferior or remain of doubtful value as far as general application is concerned. As a caution it should be stated that some, although comparatively few, of the types and general details shown are covered by patent, and are therefore not open to general use. It was not possible to obtain reliable information regarding the field so protected, useful as this would have been to the designer, because definite statements on this point could be obtained from only a few of the manufacturers.

I. Beds and Frames

Material. Mostly cast iron, for light automobile engines also aluminum alloys, and now and then cast steel or soft steel; for crank bearings of the smaller sizes nearly always brass or bronze (phosphor-bronze), for other sizes cast iron lined with white metal.

Allowable Fibre stress is comparatively low, in consideration of the importance of this machine part and the generally uncertain estimation of the stresses occurring. In the case of stationary engines this allowable fiber stress for cast iron of the best grade should not exceed 3500 lbs. per sq.in. in bending, as computed from the total explosion pressure, P_z.

The body of vertical engines is usually called the frame, that of horizontal machines usually the bed, although the use of these terms is not at all strictly defined. For both of these in the course of time fairly similar fundamental forms have been developed, which are pretty closely adhered to unless necessity demands other constructions.

In every machine part under stress it is important that the stress be taken up as far as possible in a central or axial direction. To take up the maximum total explosive stress P_z in this way is an especially important requirement in the design of beds and frames. In the case of frames this can be fulfilled easily and by simple means, in the case of engine beds, however, only by employing designs as yet little used. (See Figs. 83–86, p. 106.) The types of horizontal engine beds generally employed are all subject to considerable bending stresses. In every case the moment arms of the bending stresses, where these cannot be avoided, must be made as short as possible if a safe construction is to be obtained with the least expenditure of material. It should be considered that, with maximum total explosion pressures ranging from 300 to 400 tons, and such are found in the large engine, every $\frac{1}{16}$ of an inch off or on the length of a moment arm, has a considerable influence on the size of the bending stress. To relieve the bed or frame through connections with pedestals or foundations is of doubtful utility, since such connections are always more or less yielding, and often accompanied by other and unknown stresses. Relief in that way should never be sought and should not in any case be taken into account in strength computations. The machine bodies themselves, therefore, should under all circumstances be able to take up all stresses occurring during operation. Twisting of frame during manufacture or erection can be quite successfully prevented by sufficiently large base surfaces, structural stiffness of the casting and proper location of the foundation bolts or anchors (close to vertical walls and near the points of support).

The unit pressure or load on foundations due to weight of machine and whatever inertia forces there may be, should not exceed the following figures for the various materials employed:

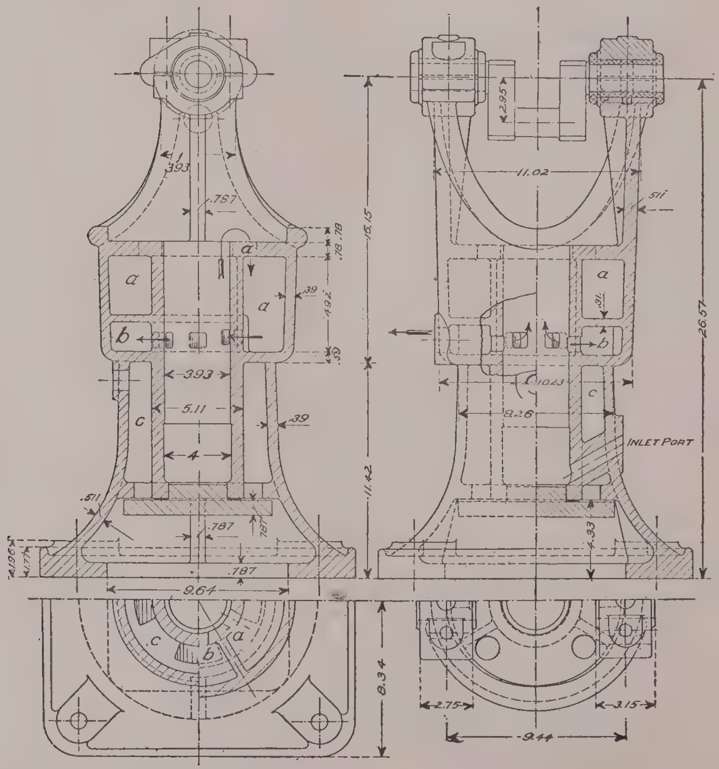
	Lbs. per sq.in.		Lbs. per sq.in.
Granite blocks	110–140	Brick and concrete	42–55
Sandstone	70– 80	Hardwood timber	55--70
Hard-pressed brick in cement	55– 70	Pine and hemlock	30–40

It is well to keep the unit load from one fourth to one third less than these figures under the crank bearings of horizontal machines. This gives a means of computing the necessary area of the supporting surfaces.

For dimensions of foundations, see p. 258.

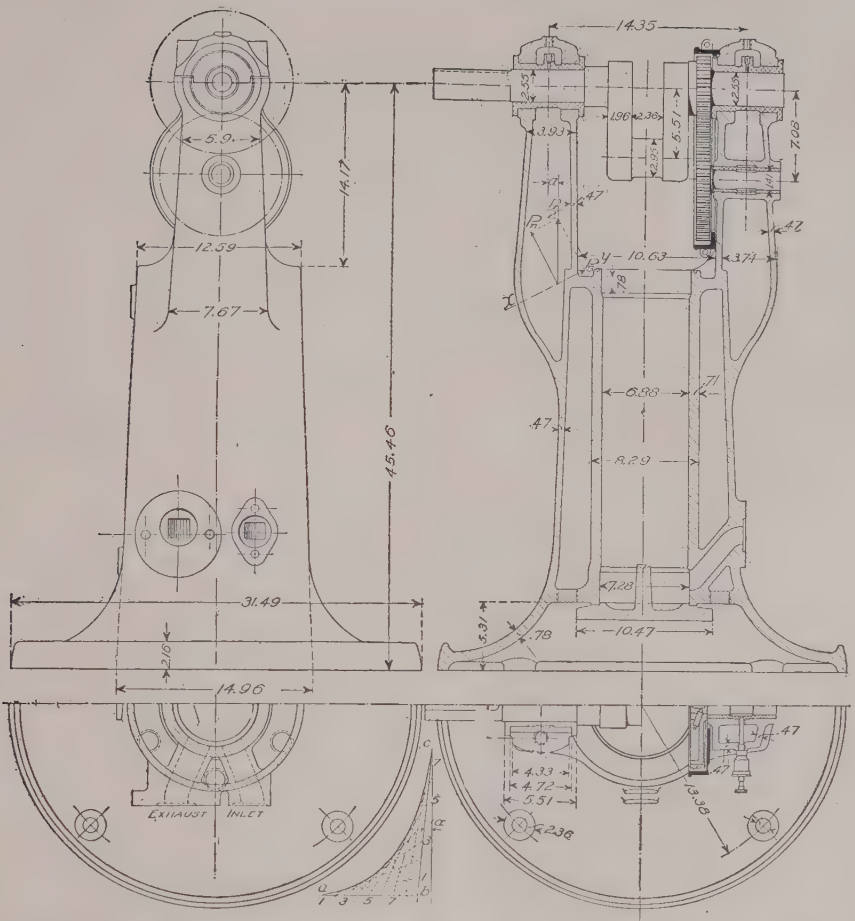
1. Frames for Vertical Engines, Crank-shaft above Cylinder (Figs. 35–43). This type of frame is to-day used only for small powers, up to 6 H.P., and for speeds not to exceed 250 r.p.m. It possesses the advantage of a fly-wheel running entirely free, and needs little or no foundation. But for larger units these frames have too little stability, owing to the high location of the gravity center, and there are other disadvantages in static respects.

Designs.



Figs. 35–38.—Frame for a 2-cycle Güldner Engine.

(The upper end of the cylinder serves as a charge pump, which, by means of a port *a*, communicates with an annular space *a* used as a receiver. The charging and discharging actions of this pump are controlled by a slide valve on the top of the upper cylinder cover, which valve is operated by the connecting rod passing through an oscillating ball joint in the valve itself. Space *b* is the exhaust chamber, *c* the jacket space. After the exhaust gases have escaped into *b* the inlet ports from *a* are opened.)



FIGS. 39–43.—Frame for a Kjelsberg Kerosene Engine, built by L. Nobel, Petersburg.

(The web projection on the cylinder head is used to prevent the piston from dropping down too far, in which case the piston rings may catch behind the counter-bore.)

Constructive Details. Length of stroke, S, generally from 1.75 to 2.0 times the cylinder diameter D, for larger sizes down to 1.5 D; height to center of main bearings from 4 to 5 S, and length of side of the generally square base from 4.5 to 5 D. Prismatic frames of this type are now used but very little, although the cost of the patterns is less and the weight of the frame 10% less than in the case of cylindrical frame bodies of equal capacity. Generally cylinder and frame are cast in one. Separate cylinder barrels cause some difficulty on account of the lateral connection with the ports, and from considerations of strength require a thickening of the walls of the frame.

Strength computation results in thickness of cylinder and jacket wall too thin for practical work; on this account with special reference to foundry practice, the dimensions may be made as follows for one-piece frames:

For D up to 5.0″, cylinder wall $(s) = .50–.60″$; frame walls $(s') = .40″$
" 7.0″, " " $(s) = .60–.70″$; " " $(s') = .45″$
" 8.0″, " " $(s) = .70–.80″$; " " $(s') = .50″$
" 9.0″, " " $(s) = .80–.90″$; " " $(s') = .55″$

With the above dimensions the tensile stress in the cylinder wall, as computed from the explosion pressure, and without reference to the influence of the frame, will be only about 1400 lbs. per sq.in. If separate cylinder barrels are used, the thickness s of the cylinder wall may be reduced to the figure found by computation [eq. (11), p. 124]. For the design of the bottom end of the cylinder see p. 134.

In the case of the larger frames, Figs. 39–43, the transition section between the bearing support and the jacket wall must be specially checked; because at this point the force $\frac{P_z}{2}$ is not taken up in an axial direction, but acts through a moment arm, a. It may be resolved into the tensile stress, P_n, and the shearing stress, P_s (see triangle of forces in Fig. 40). The stress P_s may be neglected and the computation confined to the determination of the combined stress resulting from P_n (tension) and $\frac{P_z}{2} \cdot a$ (bending). Assuming the area of the dangerous section x–y, Fig. 40, to be f sq.in., and the section modulus equal to W, we will have the maximum combined stress

$$\sigma = \frac{P_n}{f} + \frac{\frac{P_z}{2} a}{W} \text{ lbs. per sq.in.} \qquad (1)$$

The value of P_n is best found graphically. The distance a from the center line of the bearing to the neutral axis of the dangerous section, x–y, is determined most easily by balancing a stiff paper model of the section concerned. See p. 90.

Since the line of action of the fly-wheel weight G is opposite to that of the piston pressure P_z, the opposite bearing will be relieved of its load $\frac{P_z}{2}$ by a certain amount. But the weight G referred to the center line of this bearing acts as a load as soon as $\frac{P_z}{2} < G$, which condition takes place during the expansion stroke. And since this bearing is thus loaded the greater part of the time, the fly-wheel is to be kept as light as possible, and seated as closely as possible to the adjacent bearing. For details relating to the determination of bearing pressures, see under "Crank Shafts," p. 164.

The width of the jacket space should in no place be less than 1½″. The external shape of the frame usually gives a water space considerably wider than this, which is favorable to good casting. For the removal of the core sand suitable cleaning holes should be provided in the base. These are afterward closed watertight by fine threaded plugs. The closing of these core sand openings should not be made a part of the duty of the cylinder head packing. The curves at the base between jacket wall and base plate are parabolas, the construction of which is shown in Fig. 42. The usual method is to draw this curve once and then to use circular curves of approximately the same form.

The caps and bolts of the bearings must be designed for the maximum explosion pressure. The load due to the weight of the fly-wheel, added to the load on the bearing next to this wheel, can usually be neglected, because general considerations of safety and the fact that the direction of the belt pull is hardly ever definitely known, make it imperative to keep the stresses down to a comparatively low figure. Let $P = \frac{P_z}{2}$ lbs. be half of the explosion pressure, also let the distance from center to center of studs or bolts be a in. and the section modulus of the dangerous section of the bearing cap be W, then from $P=\frac{4k_bW}{a}$, in which k_b=allowable bending stress, we shall have the maximum bending stress

$$\sigma_b=\frac{aP}{4W} \text{ lbs. per sq.in.} \qquad (2)$$

If the dangerous section is in the shape of a geometrical figure, a rectangle for instance, as is usual in small machines, the section modulus W may be taken directly from a table. In other cases it is necessary to first determine the moment of inertia I of the cross-section, which is best done by dividing the given section into simple figures, and for each of these determine I referred to the gravity axis of the section. The sum of the partial moments so found is equal to the moment I of the total section. The following example makes this clear.

Example. The dangerous section of one of the bearing caps of the frame shown in Figs. 39–43 has the dimensions given in Fig. 44.[1] Since the cylinder diameter is 6.9″, the total explosion pressure P_z for a unit pressure of $p_z=255$ lbs. (for which this engine is constructed), will be $P_z=255\times .785\times 6.9^2=9500$ lbs. Of this load $\frac{9500}{2}=4750$ lbs. $=P$ will fall to each bearing. With the given distance a between bolts equal to 5.1″, the greatest bending moment in the cap is

$$M_b=\frac{P}{2}\times\frac{a}{2}=\frac{4750\times 5.1}{4}=6100 \text{ in.-lbs.}^2$$

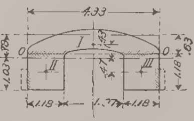


FIG. 44.—Dimensions in Inches.

For the determination of the moment of inertia the cross-section is divided into three parts, I, II, and III, Fig. 44. Minor variations and approximations in outline, such as made in the figure, are of no importance. The distance of the gravity axis from any given reference line, such as the upper horizontal line in the figure, may be computed as follows:

$$e_1=\frac{\frac{2.36\times 1.81^2}{2}+\frac{1.97\times .63^2}{2}}{(2.36\times 1.82)+(1.97\times .63)}=.78'',$$

hence $\qquad e_2=1.81-.78=1.03''.$

For sections more complex than this the position of the gravity axis is best located by cutting a proportional model out of stiff paper and balancing this on a sharp straight edge.

The several moments of inertia of the section of Fig. 44, referred to the gravity axis will be

$$\text{For area I,}\quad I_1=\frac{4.33\times .63^3}{12}+(4.33\times .63\times .43^2)\ =.593 \text{ bi-quad. in. (or in.}^4\text{).}$$

---

[1] All dimensions given in examples will be in inches, since nearly all of our strength computations are based on this unit.

[2] All results are given in round numbers, due to the fact that the slide rule is used. The approximations have no effect on the final results.

$$\text{For area II,}\quad I_{\text{II}}=\frac{1.18\times1.18^3}{12}+(1.18\times1.18\times.47^2)=.466\text{ bi-quad. in.}$$

$$\text{For area III,}\quad I_{\text{III}}=\frac{1.18\times1.18^3}{12}+(1.18\times1.18\times.47^2)=.466\text{ bi-quad. in.}$$

So that the total moment of inertia for the section will be

$$I=I_{\text{I}}+I_{\text{II}}+I_{\text{III}}=.593+.466+.466=1.525\text{ bi-quad. ins.}$$

The section modulus therefore is

$$W=\frac{I}{e_1}=\frac{1.525}{.78}=1.96,$$

and the greatest bending stress (in the top side of the section) will therefore be

$$\sigma_b=\frac{M_b}{W}=\frac{6100}{1.96}=3100\text{ lbs. per sq.in.}$$

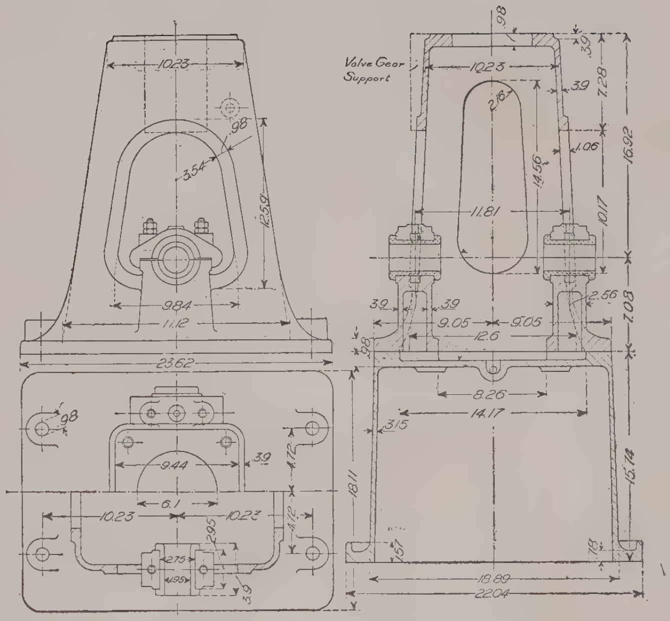
The $\frac{7}{8}''$ cap screws each have a load of $\frac{4750}{2}=2375$ lbs., which, according to column 6, of table (p. 307), is safe.

The *foundation bolts* of this type of vertical frame may have the following dimensions:

For $D=$	5.0″	7.0″	8.0″	9.0″
Diameter of bolt=	$\frac{3}{4}''$	$\frac{7}{8}''$	$\frac{7}{8}''$	1″

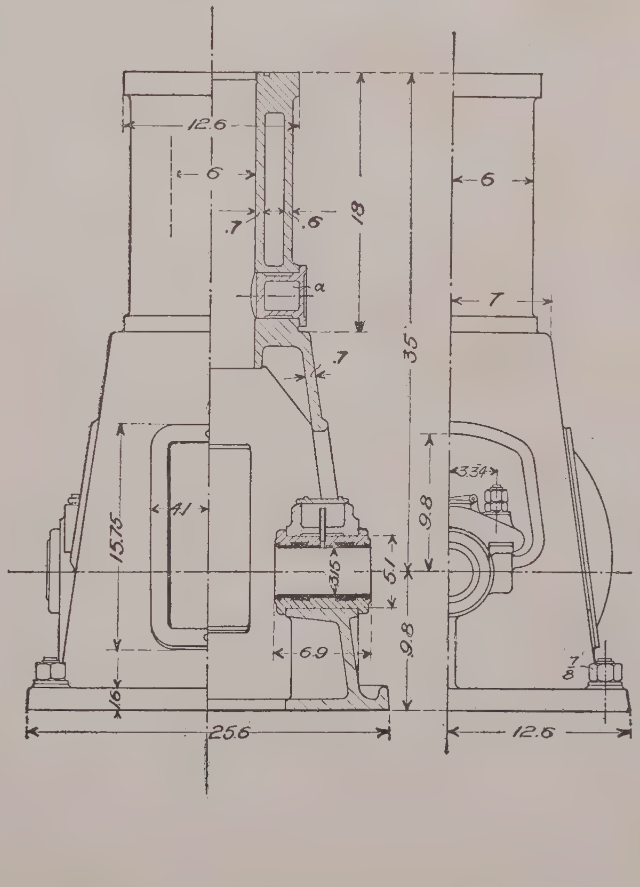
2. Standard Frames for Vertical Engines, Crank Shaft below Cylinder. The so-called box frames, Figs. 45–49, are mostly used for small engines only, although some American firms, as Westinghouse for instance, also employ them for large capacities. On account of the low position of the crank shaft this type is well adapted to high speeds of rotation and for direct connection with the power consumer. The facility with which the working parts can be tightly enclosed in such a frame make it especially suitable for use in the open or in dusty rooms. This fact has led to the development of special crank cases for automobile engines (see the various constructions shown in Part III). Such enclosed frames in themselves insure sufficient stiffness of construction, if the walls are made thick enough. In special cases, however, sufficient webbing for the side walls must be provided, especially if for purposes of lightness the crank case is made of the commonly used aluminum alloy instead of cast iron.

Design of Box Frames.



FIGS. 45–47.—Frame with Pedestal for a 2 H.P. Engine ($D=4.75''$, $S=8''$).

(The hit-and-miss governor and valve operating arrangement shown in Fig. 260, p. 206, is bolted to the left side of the frame near the top. The use of neither 2 to 1 gear nor lay shaft is necessary with this construction. Some designers use the pedestal as fuel tank or even as exhaust muffler. The latter practice is bad because of the consequent heating of the crank bearings. It is better to use the space available in the pedestal merely as a receptacle for tools or auxiliary apparatus.



FIGS. 48–49.—Frame for a Small Güldner Engine, One-piece construction ($D=6''$, $S=9''$).

(If the molding of this frame causes difficulty, a separate cylinder liner may be used, casting only the jacket wall in one piece with the rest of the frame. Hand-hole *a* is used to disconnect the wrist-pin bearing in case the piston must be removed, leaving the crank-pin bearing undisturbed. This is of importance when, on account of lack of head room, it is not easy to take out both piston and rod together. For the valve gear belonging to this model see Figs. 261 and 262, p. 206.

Constructive Details of box frames according to Figs. 45–49:

Stroke ratio $\frac{S}{D}=1$ to 1.75; distance from center of shaft to base from .7 to 1.0 S; distance center of shaft to floor from 16 to 24″ depending upon size of machine and diameter of fly-wheel. Total height of frame above base from 5.5 to 6.5 S. Length of side of square base (common construction) from 3.5 to 4.5 D. The lateral openings in the base casting should be of ample size to allow of easy dismounting or assembling of crank shaft and gear.

The thickness of the walls of the base casting is in the smaller machines of this type usually prescribed by practical considerations. Based on experience, the following proportions may be used:

For $D=5''$	$7''$	$9''$	$11''$	$13''$	$15''$
$s=\frac{1}{2}''$	$\frac{9}{16}''$	$\frac{5}{8}''$	$\frac{11}{16}''$	$\frac{3}{4}''$	$\frac{13}{16}''$

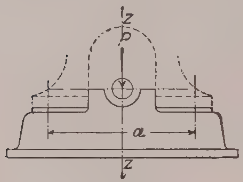


FIG. 50.

A check computation is required only to determine the tensile stress in the smallest cross-section of the frame (and of the jacket wall, if the construction is in one piece) and the bending stresses in the cross-section below the main bearing. This latter is especially necessary in case the construction is like Fig. 50, with the main bearing in a separate base plate.

Examples. 1. With reference to Fig. 50, let $P=21600$ lbs., $a=37.5''$, so that the greatest bending moment in the dangerous section Z–Z is

$$M_b=\frac{21600\times 37.5}{4}=202000 \text{ in.-lbs.}$$

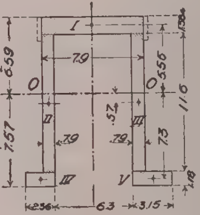


FIG 51.—Dimensions in Inches.

With reference to the neutral axis of the conventional cross-section, Fig. 51, the moment of inertia of the dangerous section will be

For area I, $$I_{\text{I}}=\frac{7.9\times 1.38^3}{12}+(7.9\times 1.38\times 5.55^2)=336.2 \text{ in.}^4;$$

For area II, $$I_{\text{II}}=\frac{.79\times 11.6^3}{12}+(.79\times 11.6\times .57^2)=105.2 \text{ in.}^4;$$

For area III, $I_{\text{III}}=$ same as $I_{\text{II}}=105.2$ in.^{4};

For area IV, $$I_{\text{IV}}=\frac{2.36\times 1.18^3}{12}+(2.36\times 1.18\times 7.0^2)=136.8 \text{ in.}^4;$$

For area V, $$I_{\text{V}}=\frac{3.15\times 1.18^3}{12}+(3.15\times 1.18\times 7.0^2)=182.4.$$

Therefore, $$I=\Sigma(I_n)=336.2+105.2+105.2+136.8+182.4=865.8 \text{ in.}^4$$

Section modulus $$W=\frac{I}{e_1}=\frac{865.8}{7.57}=114.5.$$

Consequently the maximum bending stress in the section Z–Z is

$$\sigma_b=\frac{202\,000}{114.5}=1775 \text{ lbs. per sq.in.}$$

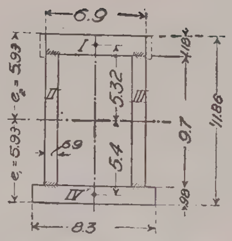


Fig. 52.—Dimensions in Inches.

2 A vertical engine, cylinder diameter 11.8″, has the dimensions given in Fig. 52 for the cross-section of the frame under the main bearings. The distance between the base bolts, see Fig. 50, is 49.3″. The normal explosion pressure, according to p. 71, will then be

$$P_z=280\,D^2 \text{ lbs.}=280\times 11.8^2=39\,000 \text{ lbs.},$$

from which the load P on each bearing will be 19500 lbs.

The greatest bending moment in the base plate below the main bearing is

$$M_b=\frac{19\,500\times 49.3}{4}=240\,000 \text{ in.-lbs.}$$

Moments of inertia of the partial areas:

$$I_{\text{I}}=\frac{6.9\times 1.18^3}{12}+(6.9\times 1.18\times 5.32^2)=230.9 \text{ in.}^4$$

$$I_{\text{II}}=\frac{.69\times 9.7^3}{12} = 52.5 \text{ in.}^4$$

$$I_{\text{III}}=\text{same as for } I_{\text{II}} = 52.5 \text{ in.}^4$$

$$I_{\text{IV}}=\frac{8.3\times .98^3}{12}+(8.3\times .98\times 5.4^2) =238.6 \text{ in.}^4$$

$$\Sigma(I_n)=574.5 \text{ in.}^4$$

Hence section modulus is $\frac{574.5}{5.93}=97.5$, and the maximum bending stress in the dangerous section is

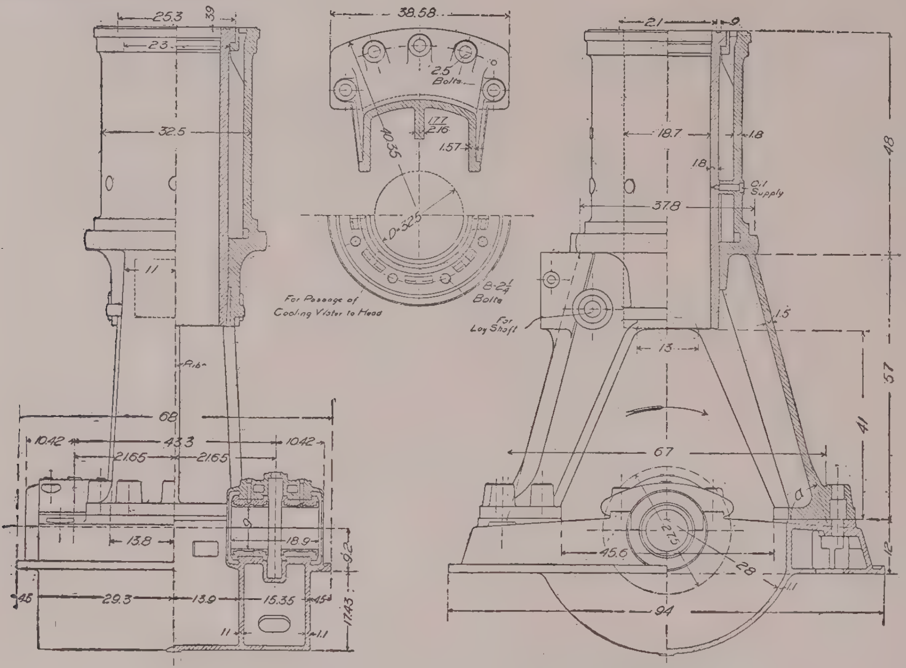
$$\sigma_b=\frac{240\,000}{97.5}=2460 \text{ lbs.}$$

The compression stress in the outer fibers above the neutral axis is neglected because the compressive strength of cast iron is much greater than its tensile strength. For further computations on base plates, see p. 97.

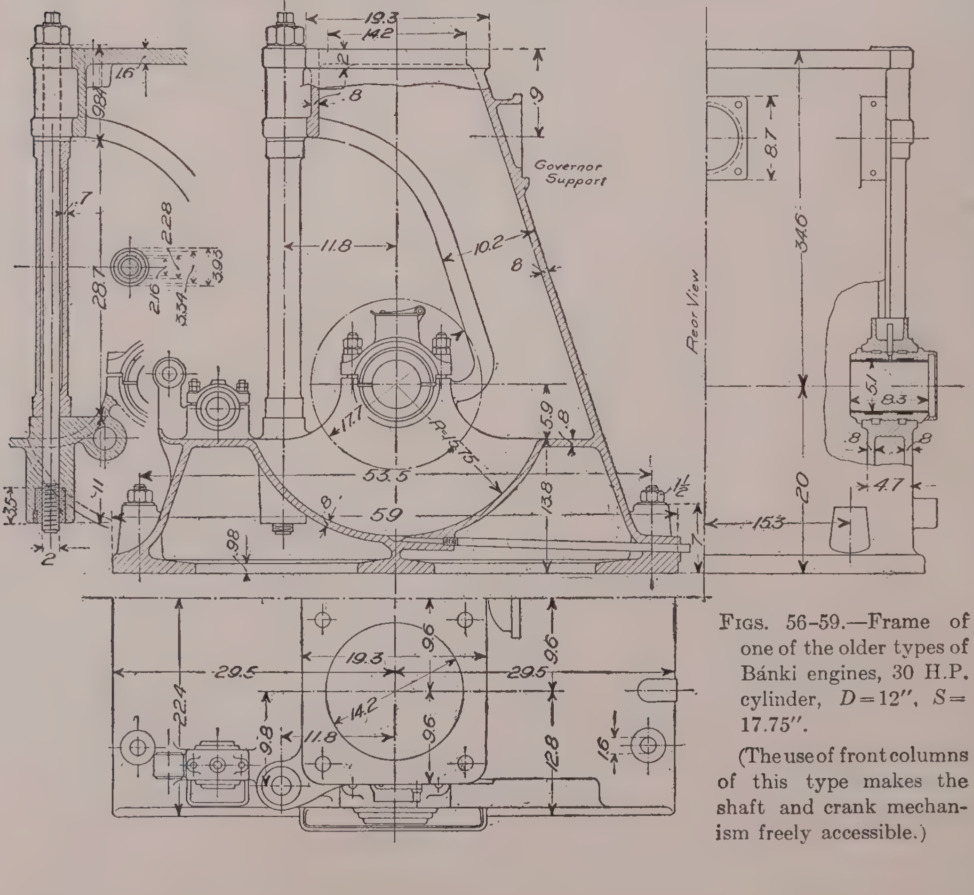
Sizes of foundation bolts should not be smaller than those given on p. 90 for frames with crank shafts above cylinders. For information regarding *foundations* see p. 257.

For medium sized vertical engines the so-called **A-frame,** borrowed from steam-engine practice, and shown in Figs. 53–55 and 60–62, has found extended application. Owing to the better disposition of material, this type of frame is considerably lighter than the box frame above discussed. Besides this the shaft and the open end of the cylinder are kept free. The substitution of separate columns (compare Figs. 56–59) for the front leg of the A-frame cannot be recommended for high-pressure engines, because this construction is less rigid and costs more.

Design of A-frames.

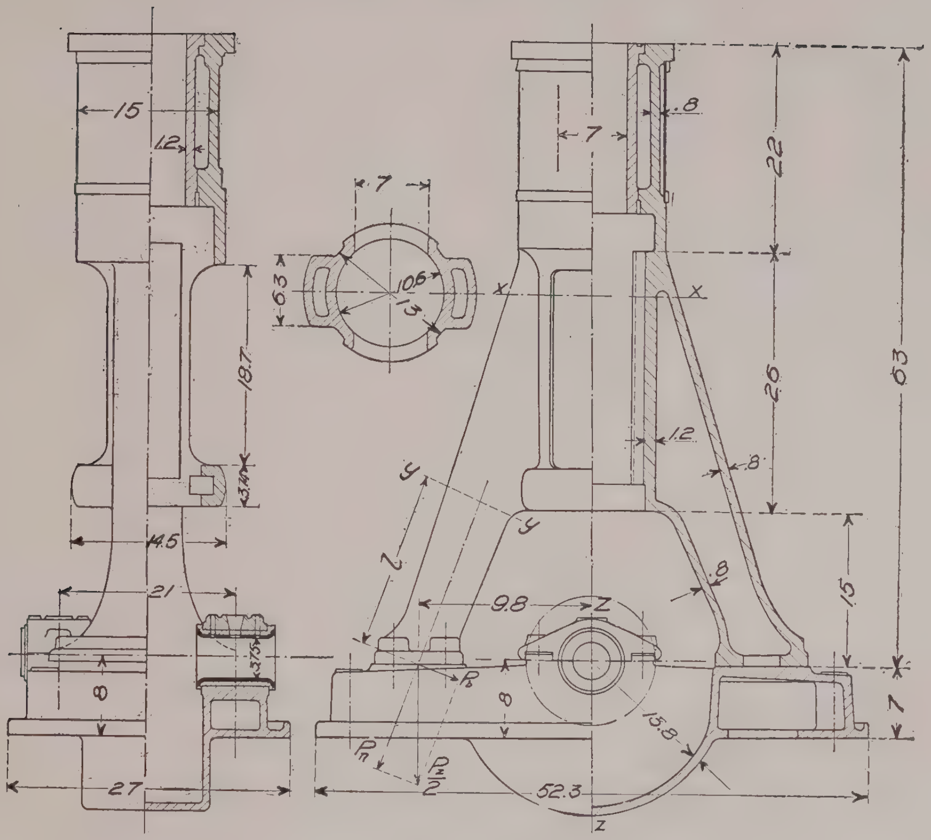


FIGS. 53–55.—Frame and Base Plate, 100 H.P. Güldner Engine.



FIGS. 56–59.—Frame of one of the older types of Bánki engines, 30 H.P. cylinder, $D=12''$, $S=17.75''$.

(The use of front columns of this type makes the shaft and crank mechanism freely accessible.)



FIGS. 60–62.—Frame with Cross-head Guides, 10 H.P. Diesel Engine of French make.

Constructive Details. Total height of machine with A-frame and cross-heads is from 8–9 S, without cross-heads from 5.5–6.5 S. Other frame dimensions approximately as given for the previous type of frame. Batter of the outside walls of the columns of the frame in the ratio of 1:4 or 1:3, depending upon the ratio $\frac{S}{D}$. That is, in Fig. 55, angle α should be in the neighborhood of 70°. A careful check computation is necessary for sections x–x (tension), y–y (tension and bending) and for the middle section z–z (bending) of the crank shaft-bearing, Fig. 62.

In the minimum horizontal section x–x, of area f sq.in., we have the tensile stress

$$\sigma=\frac{P_z}{f}\text{ lbs. per sq.in.} \qquad (3)$$

Examples. 1. Let the cross-section y–y of a frame similar to Fig. 62 have the dimensions of Fig. 63. The cylinder diameter is 10.8″, so that normal $P_z=280\,D^2=280\times10.8^2=32800$ lbs., of which one half, $P=16400$ lbs., falls to each column. At the base of the column P is resolved into the two forces P_n and P_b which put the section y–y under tension and bending.[1]

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[1] Usual assumption which gives excess safety. In fact the bending stresses will be much smaller than this on account of the rigid connection of the column with the base plate. But for closer computation satisfactory data is lacking.

The values of these forces are best determined graphically from the force parallelogram, Fig. 62. Suppose $P_n=14250$ lbs., and $P_b=5050$ lbs., and let the moment arm $l=23.6''$. Then for the section y–y of area $f=34$ sq.in. the tensile stress due to P_n will be

$$\sigma=\frac{14\,250}{34}=420 \text{ lbs. per sq.in.}$$

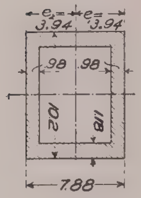

FIG. 63.—Dimensions in Inches.

The moment of inertia of the section referred to the neutral axis, Fig. 63, is

$$I=\frac{1}{12}(10.2\times 7.88^3)-\frac{1}{12}(7.84\times 5.92^3)=280 \text{ in.}^4$$

The section modulus therefore is

$$W=\frac{280}{3.94}=70.8.$$

Hence maximum bending stress

$$\sigma_b=\frac{23.6\times 5050}{70.8}=1680 \text{ lbs. per sq.in.}$$

Total maximum stress in section y–y is therefore

$$\sigma+\sigma_b=420+1680=2100 \text{ lbs. per sq.in.}^1$$

As a matter of fact the stress in the section y–y is from 3–5% smaller than this owing to the weight of cylinder and frame.

2. For a similar A-frame, the cross-section y–y of which is shown in Fig. 64, let $P_z=26400$ lbs., so that $P=13200$ lbs., and from this by means of the force parallelogram $P_n=12700$ lbs. and $P_b=3960$ lbs. The moment arm $l=21.6''$, and the area of cross-section $f=26.7$ sq.in.

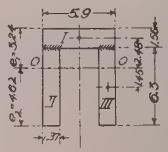

FIG. 64.—Dimensions in Inches.

$$\text{Tensile stress } \sigma=\frac{P_n}{f}=\frac{12\,700}{26.7}=475 \text{ lbs. per sq.in.}$$

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[1] Since for cast iron the allowable bending stress (K_b) is greater than the allowable tensile stress (K_z), we should for this metal strictly satisfy the requirement that

$$\beta\sigma+\sigma_b\leq K_b \quad\text{or}\quad \sigma+\frac{\sigma_b}{\beta}\leq K_z.$$

For the most common shapes of cross-section, $\beta=\frac{K_b}{K_z}=1.2$ to 1.5. With the latter value the above example would have given for the maximum total bending stress in the section the value

$$\sigma_1=(1.5\times 420)+1680=2310 \text{ lbs. per sq.in.}$$

The difference is evidently insignificant and remains so as long as σ is comparatively small as compared with σ_b. This holds good for all the examples following (except Example 3, p. 97) for which reason K_b has been assumed equal to K_z throughout, i.e., $\beta=1.0$. The inaccuracy introduced thereby increases the factor of safety as long as the relation $\sigma+\sigma_b\leq K_z$ is satisfied.

Moment of inertia of section referred to neutral axis

$$\left.\begin{aligned} I_{\text{I}} &= \frac{5.9\times1.56^3}{12}+(5.9\times1.56\times2.48^2)=58.57 \\ I_{\text{II}} \text{ and } I_{\text{III}}, \text{ each} &= \frac{1.37\times6.3^3}{12}+(1.37\times6.3\times1.45^2)=46.80 \end{aligned}\right\} \text{Total } I=152.17 \text{ in.}^4$$

$$\text{Modulus of section } W=\frac{152.17}{3.24}=47.0.$$

$$\text{Bending moment } M_b=21.6\times3960=85500 \text{ in.-lbs.}$$

so that the maximum bending stress, in the external fibers of the column

$$\sigma_b=\frac{85500}{47}=1830 \text{ lbs. per sq.in.}$$

Hence total stress is $\sigma+\sigma_b=475+1830=2305$ lbs. per sq.in.

In determining the stresses in the section z–z of the base plate, which section is usually under the greatest stress, the stiffening due to the foundation is neglected because its effect is uncertain, and counterbalanced by the weight of fly wheel and crank shaft, by unknown accidental stresses and by the drawing up of the foundation bolts. For the same reasons any decreasing effect that the dead weight of frame and cylinder may have on the value of P_z is also neglected. If the center line of the cylinder bisects the distance between the middle planes of the main bearings, the load on each bearing will be simply $P=\frac{P_z}{2}$; in other cases P, referred to the middle of the bearing, can be easily determined.

Example. 3. The cross-section z–z, through the support of a main bearing having ring oilers, Fig. 65, is to take care of a load of $P=55000$ lbs. Let the distance between the column bolts on each side of the bearing be $a=55.2''$, then the maximum bending moment will be

$$M_b=\tfrac{1}{4}Pa=\frac{55000\times55.2}{4}=762000 \text{ in.-lbs.}$$

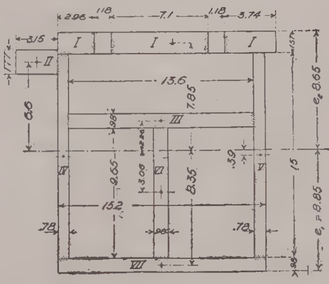

FIG. 65.—Dimensions in Inches.

The total moment of inertia I is equal to $\Sigma(I_n)$, Fig. 65.

$$I=I_{\text{I}}+I_{\text{II}}+I_{\text{III}}+I_{\text{IV}}+I_{\text{V}}+I_{\text{VI}}+I_{\text{VII}};$$

hence

$$I=\left[\frac{13.80\times1.57^3}{12}+13.80\times1.57\times7.85^2\right]+\left[\frac{3.15\times1.77^3}{12}+3.15\times1.77\times6.6^2\right]$$

$$+\left[\frac{13.64\times.98^3}{12}+13.64\times.98\times2.26^2\right]+2\left[\frac{.78\times15^3}{12}+.78\times15\times.39^2\right]$$

$$+\left[\frac{.98\times9.65^3}{12}+.98\times9.65\times3.06^2\right]+\left[\frac{15.2\times.98^3}{12}+15.2\times.98\times8.35^2\right]$$

$$=1334.4+243.6+69.1+(2\times220.8)+161.7+1041.2=3291.6 \text{ in.}^4$$

Therefore the modulus is $$W=\frac{I}{e_1}=\frac{3291.6}{8.85}=371.0,$$

and the maximum bending stress in the underside is

$$\sigma_b=\frac{M_b}{W}=\frac{762\,000}{371}=2060 \text{ lbs. per sq.in.}$$

In the examples so far considered it was assumed that the base plate was a frame open in the center, i.e., consisting of the lateral beams connected across the ends by two other beams carrying the bearings. This is the usual construction for small and medium-sized machines. In large-size vertical machines this base-plate frame is usually closed on the under side by giving it a curved bottom to fit the crank circle. In that way the four sides of the base plate are connected in such a way as to materially increase the strength and stiffness of the entire plate. This is shown more clearly by the following example, which shows a computation not only for the section of the plate directly under the bearing, but for the entire cross-section. The fact that the bed plate is not symmetrical with reference to the cylinder axis, compels us to consider the entire section. Had it been symmetrical, only one half of the section would need to be considered.

Example. 4. Cylinder diameter $D=19.7''$, maximum total pressure on piston $=280\times19.7^2=109\,000$ lbs. Distance between points of application of force (see Fig. 50), $a=49.3''$. Bending moment due to the total piston load (not half this load, since the entire cross-section is considered) is

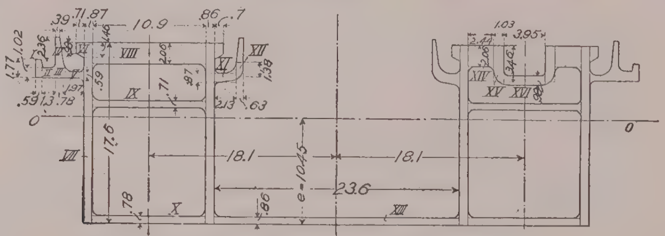

FIG. 66.—Dimensions in Inches.

$$M_b=\frac{109\,000\times49.3}{4}=1\,338\,000 \text{ in.-lbs.}$$

The moment of inertia of the cross-section is to be determined exactly according to the dimensions of the actual construction shown in Fig. 66. This makes it necessary to subdivide the section into a number of partial areas. Many of these partial areas, as I, IV, and VI, could without great error be entirely neglected, others with minor changes could be much simplified. Nothing of this kind has been done in order to show the complete method of computation for a definitely given cross-sectional area.

Cross-section through left bearing:

$$I_{\text{I}}=\frac{.59\times1.77^3}{12}+(.59\times1.77\times4.65^2)=.27+22.6=22.87 \text{ in.}^4$$

$$I_{\text{II}}=\frac{1.3\times1.02^3}{12}+(1.3\times1.02\times4.25^2)=.12+24.0=24.12 \text{ in.}^4$$

$$I_{\text{III}}=\frac{.78\times1.58^3}{12}+.78\times1.58\times4.57^2)=.26+25.7\qquad=25.96\text{ in.}^4$$

$$I_{\text{IV}}=2\left[\frac{\frac{.39+.78}{2}\times2.36^3}{12}+\left(\frac{.39+.78}{2}\times2.36\times6.4^2\right)\right]=2(.64+57.0)\qquad=115.28\text{ in.}^4$$

$$I_{\text{V}}=\frac{1.97\times.59^3}{12}+(1.97\times.59\times4.05^2)=.034+19.10\qquad=19.13\text{ in.}^4$$

$$I_{\text{VI}}=2\left[\frac{.71\times\left(\frac{.98+1.46}{2}\right)^3}{12}+\left(.71\times\frac{.98+1.46}{2}\times6.5^2\right)\right]=2(.11+36.6)=73.42\text{ in.}^4$$

$$I_{\text{VII}}=2\left[\frac{.87\times17.6^3}{12}+(.87\times17.6\times1.65^2)\right]=2(395+41.8)\qquad=873.60\text{ in.}^4$$

$$I_{\text{VIII}}=\frac{10.9\times2.06^3}{12}+(10.9\times2.06\times6.1^2)=7.7+829.0\qquad=836.70\text{ in.}^4$$

$$I_{\text{IX}}=\frac{10.9\times.71^3}{12}+(10.9\times.71\times1.26^2)=.32+12.3\qquad=12.62\text{ in.}^4$$

$$I_{\text{X}}=\frac{10.9\times.78^3}{12}+(10.9\times.78\times10.1^2)=.51+862\qquad=862.51\text{ in.}^4$$

$$I_{\text{XI}}=\frac{2.13\times.87^3}{12}+(2.13\times.87\times3.96^2)=.12+29.3\qquad=29.42\text{ in.}^4$$

$$I_{\text{XII}}=\frac{.63\times1.38^3}{12}+(.63\times1.38\times4.65^2)=.13+18.8\qquad=18.93\text{ in.}^4$$

$$\Sigma I_l=2914.56\text{ in.}^4$$

For the crank case

$$I_{\text{XIII}}=\frac{23.6\times.86^3}{12}+23.6\times.86\times10^2=1.34+2030=2031.34.$$

Cross-section through right main bearing.

On this side the areas XIV, XV, and XVI appear instead of the area VIII of the left section. All other areas are exactly the same on both sides. Hence for the right section we have simply

$$I_{\text{XIV}}=2\left[\frac{2.44\times2.06^3}{12}+(2.44\times2.06\times6.1^2)\right]=2(1.78+188.0)\qquad=379.56$$

$$I_{\text{XV}}=2\left[\frac{1.03\times3.46^3}{12}+(1.03\times3.46\times5.4^2)\right]=2(3.54\times104.0)\qquad=215.08$$

$$I_{\text{XVI}}=\frac{3.95\times.98^3}{12}+(3.95\times.98\times3.86^2)=.31+58.0\qquad=58.31$$

$$\text{Total}\quad=652.95$$

Hence moment of inertia of right cross-section

$$I_r = 2914.56 - 835.70 + 652.95 = 2730.81 \text{ in.}^4$$

which is 183.85 in.^{4} smaller than for the left side.

From the partial moments of inertia the total moment for the entire section is

$$I = I_l + I_{\text{XIII}} + I_r = 2914.56 + 2031.34 + 2730.81 = 7676.71 \text{ in.}^4$$

Since the distance from the neutral axis to the fiber under greatest stress is $e = 10.45''$, the section modulus will be

$$W = \frac{7676.71}{10.45} = 735,$$

and finally, from the value for M_b above computed, the maximum fiber stress is

$$\sigma_b = \frac{1\,338\,000}{735} = 1820 \text{ lbs. per sq.in.}$$

The moment of inertia of the crank case amounts to about 35% of the sum of the moments of the two bearing cross-sections. If these last two are considered alone, the total section modulus will be only about 484, since the fiber distance is increased to about $e = 11.8''$, leaving out the crank-case wall. It is evident, therefore, that this wall, or the lower surface of the base plate, increases the strength of the casting very materially.

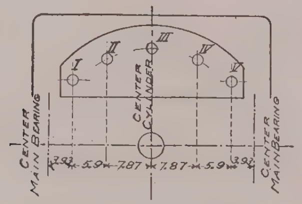

FIG. 67.—Dimensions in Inches.

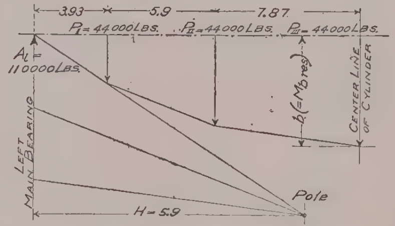

FIG. 68.—Dimensions in Inches.

As a general thing, the fiber stress in the middle vertical section of the base plate perpendicular to the axis of the crank shaft is less than in the vertical sections through the axis of the bearings. Under certain circumstances, however, as for instance great distance between bearings, bad connections between frame columns and the plate, the reverse condition may exist, so that a check computation should be made. The following example shows the method.

Example. 5. Suppose a base plate similar to the one preceding is to stand a piston load of $P_z = 220\,000$ lbs., corresponding to 110000 lbs. pressure on each bearing. Let the distance between middle sections of bearings be 35.5″. The bolts connecting columns and plate are distributed in five groups, according to Fig. 67, so that each group is loaded with $\dfrac{220\,000}{5} = 44\,000$ lbs.

The first thing to do is to determine the resulting bending moment due to P_z distributed in the planes marked I to V. The best means of doing this is the force polygon. Since the component forces are of equal magnitude and are distributed symmetrically about the middle plane III, the resultant force will be in plane III, and it is necessary to consider or make the computation for only one half of the plate, as is done in Fig. 68. From this the maximum bending moment $M_{b\ \mathrm{res.}}=1\,005\,000$ in.-lbs. (In explanation, $M_{b\ \mathrm{res.}}=HbKl$ in in.-lbs., when H = polar distance in the force polygon, b is the maximum distance cut off by the two closing lines of the force polygon on the line of the resultant force, K = scale on which the forces are represented, and l = scale of length. Fig. 68 shows the force polygon one fourth of the original size. In this original drawing $H=5.90''$, $b=2.44''$, $K=27\,940$ lbs. per in., and $l=2.5''$ to the inch. Hence

$$M_{b\ \mathrm{res.}}=5.90\times2.44\times27940\times2.5$$

$$=1\,005\,000 \text{ in.-lbs.,}$$

as given above.)

For the determination of the section modulus of that section of the base plate under greatest stress, perpendicular to the crank shaft, the dimensions are given in Fig. 69. As indicated there, the total area must again be divided into a number of partial areas. The left half of the section, not shown, is the same as the right. As before

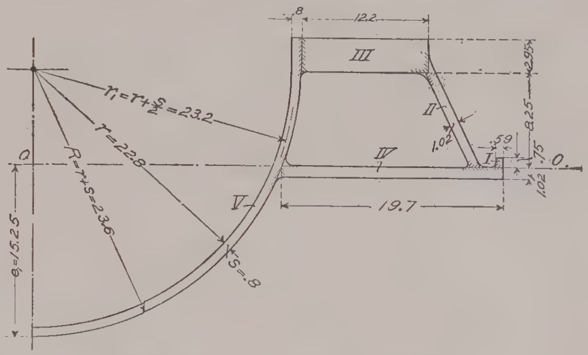

FIG. 69.—Dimensions in Inches.

$$I_{\mathrm{I}}=2\left[\frac{.59\times.75^3}{12}+(.59\times.75\times.37^2)\right]=2(.0210+.062)=1.66 \text{ in.}^4$$

$$I_{\mathrm{II}}=2\left[\frac{1.02\times8.25^3}{12}+(1.02\times8.25\times4.13^2)\right]=2(47.7+144)=383.40 \text{ in.}^4$$

$$I_{\mathrm{III}}=2\left[\frac{12.2\times2.95^3}{12}+(12.2\times2.95\times9.85^2)\right]=2(26.1+3490)=7032.20 \text{ in.}^4$$

$$I_{\mathrm{IV}}=2\left[\frac{19.7\times1.02^3}{12}+(19.7\times1.02\times.51^2)\right]=2(1.75+5.27)=14.04 \text{ in.}^4$$

For area V, considered as half of a circular ring, the general formula for the moment of inertia about its own neutral axis is

$$I=.1098(R^4-r^4)-\frac{.283R^2r^2(R-r)}{R+r},$$

and, with the dimensions of Fig. 69, we shall have

$$I=.1098(23.6^4-22.8^4)-\frac{.238\times23.6^2\times22.8^2(23.6-22.8)}{23.6+22.8}=3090 \text{ in.}^4$$

The distance from the center of the semicircular ring to its neutral axis is

$$=\frac{4}{3\pi}\left(\frac{R^3-r^3}{R^2-r^2}\right)=\frac{4}{3\pi}\left(\frac{23.6^3-22.8^3}{23.6^2-22.8^2}\right)=16.85''.$$

The position of this neutral axis is therefore

$$16.85-(23.6-15.25)=8.5''$$

below the neutral axis $o-o$ of the section in Fig. 69. Hence the moment of inertia of area V with reference to axis $o-o$ is increased to

$$I_V = 3090 + 8.5^2 \times \frac{\pi}{2} \times (23.6^2 - 22.8^2) = 3090 + 3540 = 6630 \text{ in}^4.$$

The sum of the partial moments gives the total moment of inertia for the section

$$I = 1.66 + 383.40 + 7032.20 + 14.04 + 6630 = 14061.3 \text{ in}^4.$$

Since e_1 the maximum fiber distance, $= 15.25''$, the section modulus is

$$W = \frac{14061.3}{15.25} = 922.0.$$

The bending moment, $M_{b\,\text{res.}} = 1\,005\,000$ in.-lbs., above determined, therefore, gives a maximum fiber stress in the section examined of

$$\frac{1\,005\,000}{922.0} = 1090 \text{ lbs. per sq.in.}$$

It is clear from the above that in the central plane considered the stresses may under certain circumstances be quite considerable.

3. Engine Beds. The old type engine beds with overhung cylinders are being replaced more and more by a type of frame in which the cylinder is entirely, or at least partly, supported by the frame. For equal weight this furnishes a construction of much greater stiffness and strength with pleasing form. For that reason these frames are now also used extensively for small machines. For large machines the two side walls of the frame are usually connected by a crank-case wall across the bottom, the whole being cast in one piece. In the very largest models, however, the side walls are often separately cast.

Single-beam frames, so-called bayonet frames, which as Corliss or Tangye frames have found application up to the highest powers in steam-engine practice, have not shown themselves well adapted to gas-engine construction. Delamare and Deboutteville tried this type of engine bed in some of the first forms of their large simplex engines, but were compelled to abandon it very soon. The reasons for their failure are not far to seek. In a Corliss frame, the main crank-shaft bearing must take care of the entire piston load, and since this load acts at an arm equal to the distance between center line of cylinder and center line of bearing, the bearing pressures are nearly twice as great as in the case of other frames. On the other hand, the double beam, or center crank type of frame, distributes this load between two bearings, and since, in this case, the moment arm above mentioned does not exist, the load on each bearing is simply one-half of the total piston load. Further, the Corliss frame, owing to the manner in which the load on the piston is taken up, is subject to a considerable bending moment in the horizontal plane. This is not found in the ordinary type of frame. In Corliss frames, therefore, not only the crank-shaft itself, but also the frame parts must be made considerably heavier. These and other disadvantages, make this frame unsuited for, or at least little adapted to, the larger sizes of gas engines. This is apparently contrary to the experience of steam-engine designers, which, however, owing to the different conditions, has no bearing on this case.[1]

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[1] TRANSLATOR'S NOTE. To judge from present American practice with regard to large engines, the difficulties mentioned have been largely overcome.

Constructive Details. The height of center of cylinder and bearings above the floor should, on account of the work of starting, lubricating, etc., not be less than about 28″ in the smaller engines; in large machines, from consideration of strength mainly, it should not exceed from 48–60″. For the beds of the smaller machines, the general shape of the casting and the requirements of foundry practice usually call for a minimum thickness of the walls of the hollow casting such that stresses exceeding the allowable hardly ever occur. Of so much the greater importance, however, is the careful design of large frames if sufficient strength and stiffness are to be obtained in suitable form and without waste of material.

The dangerous section in horizontal engine beds, the section which must be most carefully considered in design, is usually found in a plane perpendicular to the cylinder axis, and located somewhere between the main bearings and the seat of the cylinder, see Fig. 70. In this plane, the bending moment, whose arm is the distance from the center line of the cylinder to the neutral axis of the frame section under consideration, and which moment adds itself to the tensile stress already acting on this section, reaches its maximum. Other parts of the frame under severe stress are the vertical sections through the middle planes of the main bearings, especially on the fly-wheel side, where, besides the piston load, gravity forces also come into action. These, together with the cylinder flange, all require careful checking. An increase in the capacity of an engine not only calls for an increased cylinder diameter, but with it also increase the height of the frame and the magnitude of the bending moment above mentioned. For this reason the difficulties encountered in economic but safe frame design, increase very rapidly as the upper limits of capacity are approached, especially since at the same time perfection of workmanship in the shop grows less certain. The constructive means which the designer may employ in such cases will be shown in later examples.

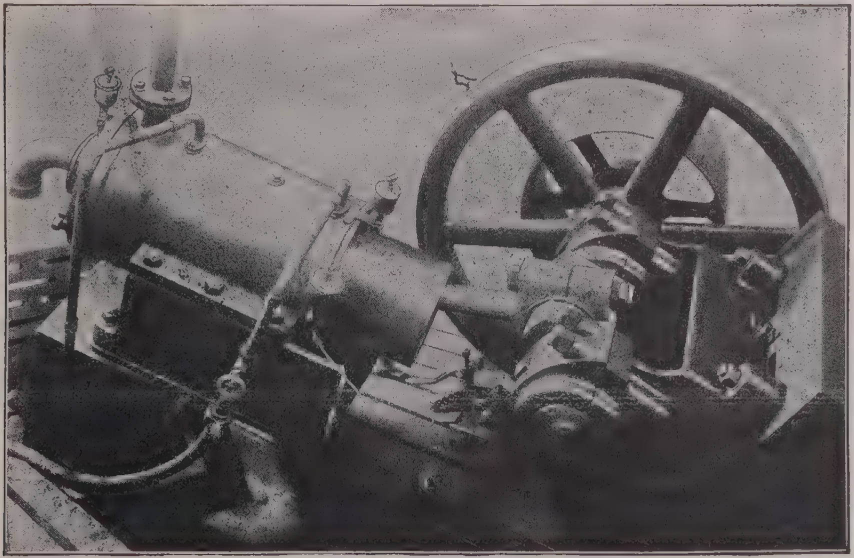
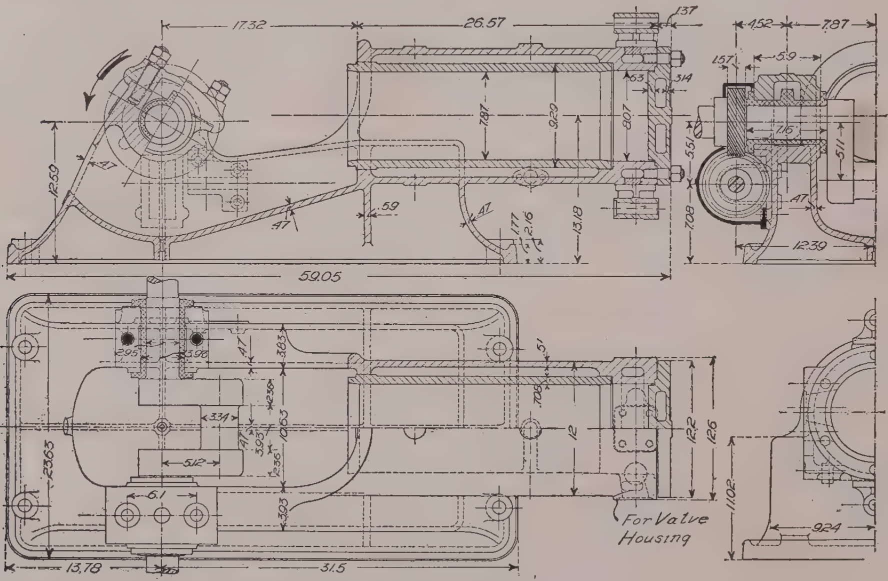


Fig. 70.—Result of Poor Frame Construction.

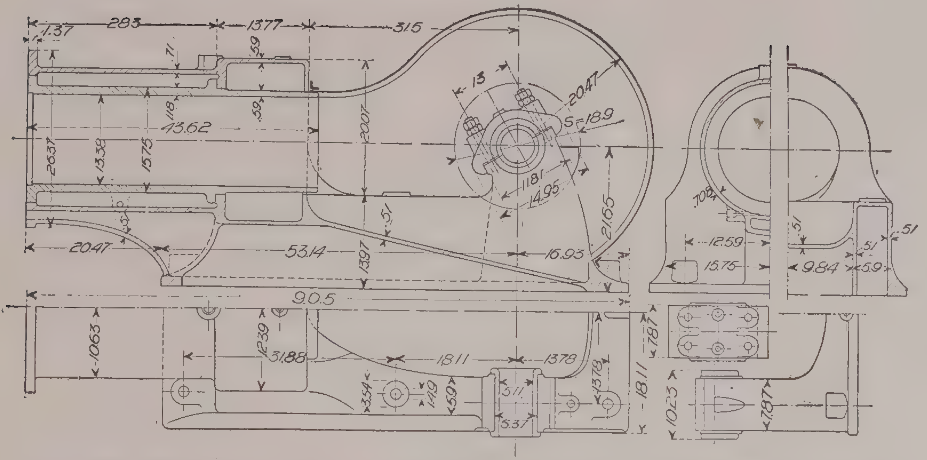
The necessity of keeping the most important bending moment with which we have to deal in the design of a frame down to its lowest possible figure, calls for a design of the vertical cross-section concerned, such that the neutral axis of this section shall be as close as possible to the center line of the cylinder. It is radically wrong, therefore, to mass great bodies of metal near the bottom of the section; the metal is needed in the upper parts of the section. Fig. 70 shows a good example of what is likely to happen when this simple rule is neglected. Further, the height of the crank bearings should be kept as low as possible, while it is good design to carry the lateral walls as high as they can be made. For small engines, these walls should extend at least to the horizontal center plane of the cylinder; for large engines they should go beyond this plane—the higher, the better. It is a costly mistake, and not one pleasing to the eye, to keep the side walls of the frame much below the level of the crank bearings, although even to-day this is quite often done. Certain forms of main bearings (as for instance, those shown in Figs. 71–80, inclined inward) compel this statically wrong construction of the side walls of frames, and their use is, therefore, permissible only in the smaller sizes.

The most central taking up of the force of the explosion in the cylinder is a necessity in gas-engine construction, and this is a fundamental rule which can not be too often repeated. Its observance even in the design of small engines is of advantage. Unfortunately, the rule can only be partially followed in the usual types of frames. Even in the most favorable cases, the distance between the neutral axis of the dangerous section and the center line of the cylinder is sufficiently great to result in a bending stress approximately equal to the tensile stress on the section, so that the total stress is nearly double what it would be if the load P_z could be balanced exactly in a central line. If the common type of frame is abandoned, an approximately central distribution of forces can be obtained most easily by putting in, above the horizontal central plane, a rigid connection between the main bearings and the seat of the cylinder. See Figs. 83–86. This idea was already carried out in the older Simplex engines; the latest models still retain it, and rightly so.

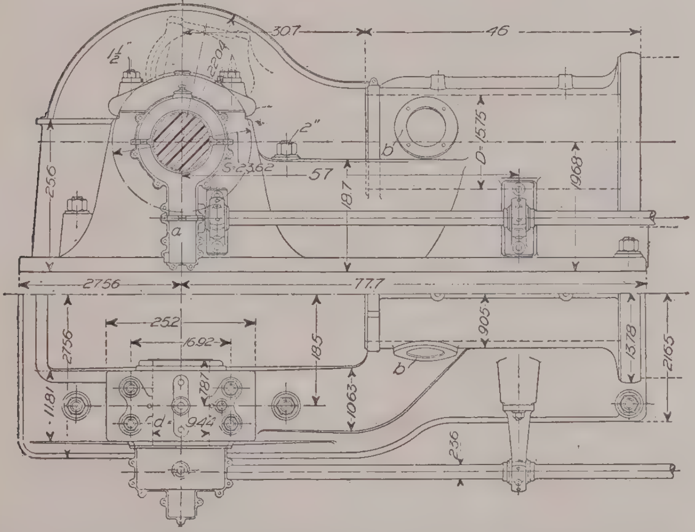
Designs of Horizontal Frames.



Figs. 71–74.—Frame for a 6 H.P. Gas Engine.



FIGS. 75–80.—Frame for a 25 H.P. Wenzel Engine, built by F. Zimmerman & Co., Halle S.



FIGS. 81 and 82.—Frame with Cross-head Guide for a Diesel-Güldner Engine.

The lower part of the bored guide is water cooled throughout the contact length. For details of the main bearing, see Figs. 109 and 110, p. 115. (The gear case *a* communicates with the oil chamber of the main bearing in order to increase both the quantity of the oil in circulation and the external cooling surfaces. After a slight turning of the cross-head, the wrist-pin may be loosened through the opening *b*.)

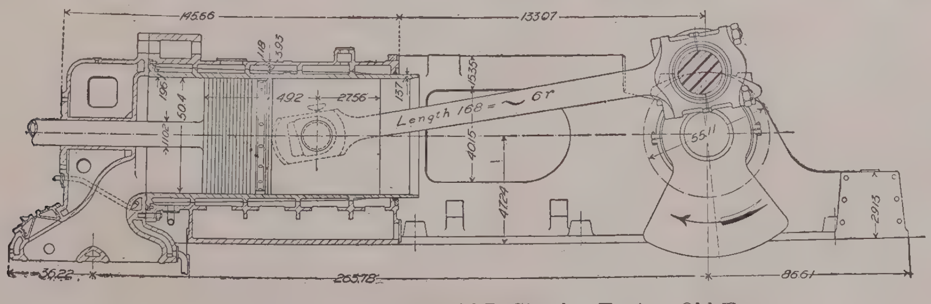
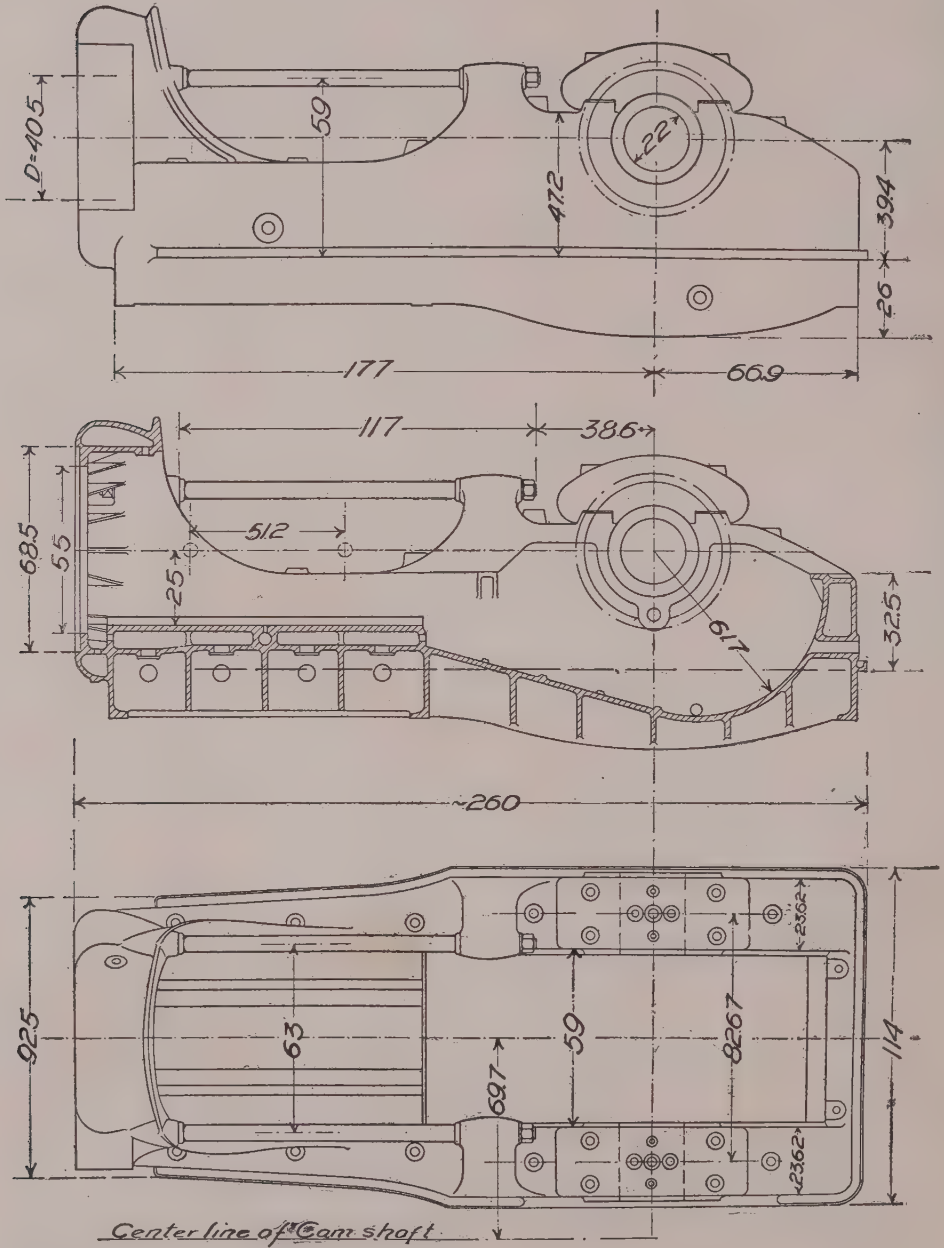
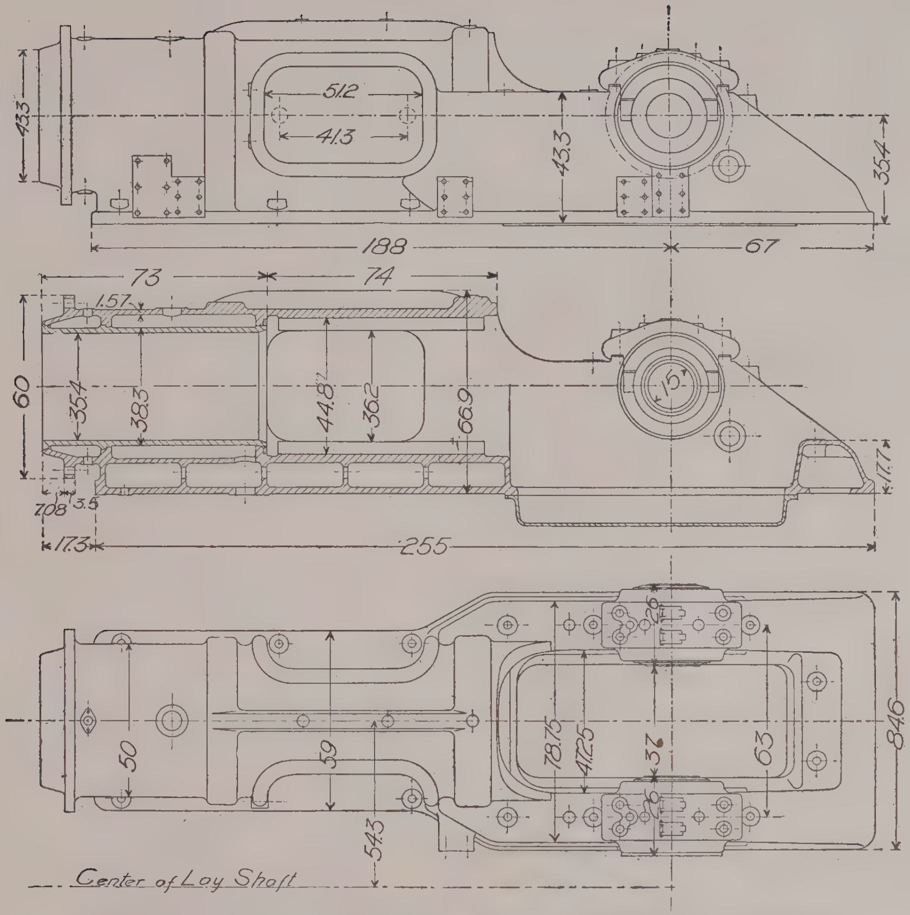


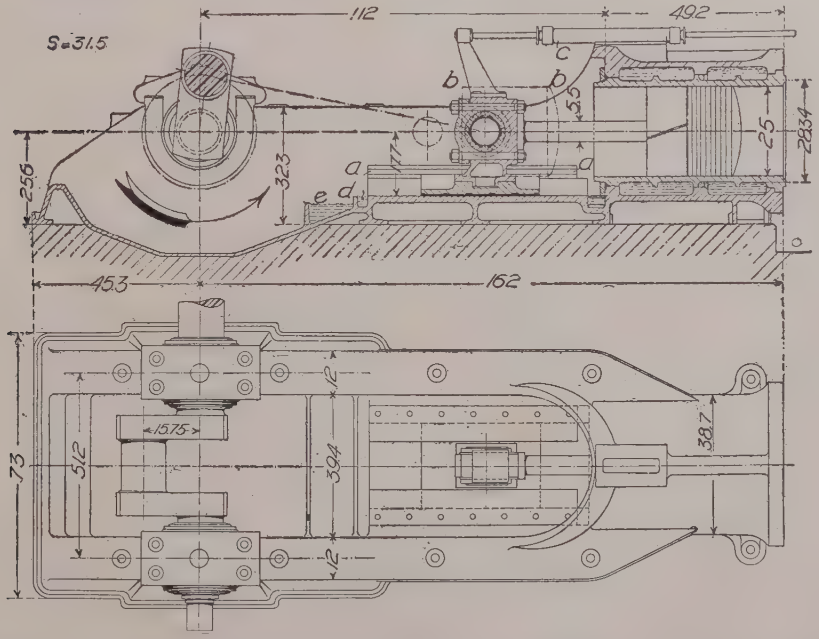
FIG. 83.—Frame for a 700–800 H.P. Simplex Engine, Old Type.



FIGS. 84–86.—Frame for a 1000 H.P. Double Acting Nürnberg Gas Engine.



FIGS. 87-89.—Frame for a 300 H.P. Gas Engine. L. Soest & Co., Düsseldorff.



FIGS. 90 and 91.—Frame for one of the Older Type of Nürnberg Gas Engines, 160 H.P.

(After the piston rod is removed, the piston may be pulled forward to the position *b*, resting on the ledges *a*. The piston is then easily taken out without disturbing the connecting rod. *c* is a stuffing-box in the pipe system for cooling the piston. The cylinder-oil drip gathers at *d* and *e*. The level in *d* is maintained to the same point, the oil serving to lubricate the slipper cross-head.)

Examples. 1. The dangerous section in the frame shown in Fig. 92 is to resist a maximum load of 17600 lbs. The area of the section = 37.3 sq.in., hence the tensile stress in the section is

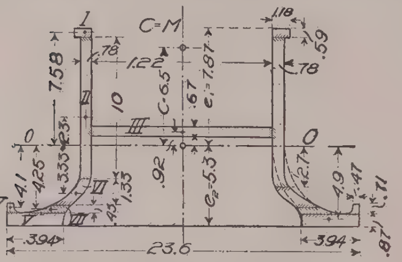


FIG. 92.—Dimensions in Inches.

$$\sigma = \frac{P_z}{f} = \frac{17600}{37.3} = 470 \text{ lbs. per sq.in.}$$

As before, the total moment of inertia of the section is found from the moments of the partial areas as follows:

$$I_{\text{I}} = \frac{1.18 \times .59^3}{12} + (1.18 \times .59 \times 7.58^2) = 40.02 \text{ in.}^4$$

$$I_{\text{II}} = \frac{.78 \times 10^3}{12} + (.78 \times 10 \times 2.3^2) = 106.30 \text{ in.}^4$$

$$I_{\text{III}} = \frac{6.10 \times .67^3}{12} + (6.1 \times .67 \times .92^2) = 3.59 \text{ in.}^4$$

$$I_{\text{IV}} = \frac{.47 \times .71^3}{12} + (.47 \times .71 \times 4.1^2) = 5.61 \text{ in.}^4$$

$$I_{\text{V}} = \frac{3.94 \times .87^3}{12} + (3.94 \times .87 \times 4.9^2) = 82.21 \text{ in.}^4$$

$$I_{\text{VI}} = \frac{.87 \times 1.33^3}{12} + (.87 \times 1.33 \times 3.33^2) = 12.97 \text{ in.}^4$$

$$I_{\text{VII}} = \frac{1.57 \times .43^3}{12} + (1.57 \times .43 \times 4.25^2) = 12.21 \text{ in.}^4$$

For $\frac{1}{2}$ of the cross-section, $\Sigma(I_n) = 262.91 \text{ in.}^4$

and since $e_1 = 7.87''$, the section modulus is

$$W = \frac{2 \times 262.91}{7.87} = 66.8.$$

Distance between neutral axis of section and center line of cylinder $(CM) = c = 6.5''$. Hence maximum bending moment

$$M_b = 6.5 \times 17600 = 114300 \text{ in.-lbs.}$$

and therefore bending stress in the outer fiber is

$$\sigma_b = \frac{114300}{66.8} = 1710 \text{ lbs. per sq.in.}$$

The maximum total stress in the fiber is consequently

$$\sigma + \sigma_b = 470 + 1710 = 2180 \text{ lbs. per sq.in.}$$

In spite of the fact that in this case the frame walls extend $1\frac{3}{8}''$ above the center line of the cylinder, the bending stress is more than $3\frac{1}{2}$ times the tensile stress.

2. The frame section under greatest stress, shown in Fig. 93, is for an engine for which $D=13.75''$; the total area of the section is 107 sq.in. At the inner dead center position of the piston the load will be $P_z=280\,D^2=280\times13.75^2=$ about 53000 lbs. The tensile stress in the section will be

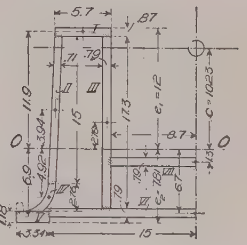


FIG. 93.—Dimensions in Inches.

$$\sigma=\frac{53000}{107}=495 \text{ lbs. per sq.in.}$$

The bending moment, due to P_z and the arm $c=10.2''$, is

$$M_b=53000\times10.2=540\,000 \text{ in.-lbs.}$$

Moments of inertia of the partial areas, referred to the neutral axis:

$$\left.\begin{aligned}
I_{I}&=\frac{5.7\times.87^3}{12}+(5.7\times.87\times11.9^2) &&= 703.3 \text{ in.}^4\\
I_{II}&=\frac{.71\times15^3}{12}+(.71\times15\times3.94^2) &&= 365.0 \text{ in.}^4\\
I_{III}&=\frac{.79\times17.3^3}{12}+(.79\times17.3\times2.76^2) &&= 443.0 \text{ in.}^4\\
I_{IV}&=\frac{.75\times2.76^3}{12}+(.75\times2.76\times4.92^2) &&= 51.3 \text{ in.}^4\\
I_{V}&=\frac{3.34\times1.18^3}{12}+(3.34\times1.18\times6.9^2) &&= 188.5 \text{ in.}^4\\
I_{VI}&=\frac{15\times.79^3}{12}+(15\times.79\times6.7^2) &&= 530.6 \text{ in.}^4\\
I_{VII}&=\frac{8.7\times.79^3}{12}+(8.7\times.79\times1.3^2) &&= 11.9 \text{ in.}^4\\
&\text{Sum of moments for half section}=2293.6 \text{ in.}^4
\end{aligned}\right\}
\begin{array}{l}\text{Moment of inertia for}\\ \text{entire section}\\ =2\times2293.6=4587.2 \text{ in}^4.\end{array}$$

Since $e_1=12.0''$, the section modulus is

$$W=\frac{4587.2}{12}=383.0$$

and the greatest bending stress in the upper fiber will be

$$\sigma_b=\frac{540\,000}{383}=1410 \text{ lbs. per sq.in.,}$$

and hence total stress in the outer fiber of the section is

$$\sigma+\sigma_b=495+1410=1905 \text{ lbs. per sq.in.}$$

3. For a Nürnberg engine, cylinder diameter=35.4″, the section under greatest stress in each one of the beams forming the side walls of the frame is shown in Fig. 94. The explosion pressure is as high as 450 lbs., so that the maximum load on each beam will be

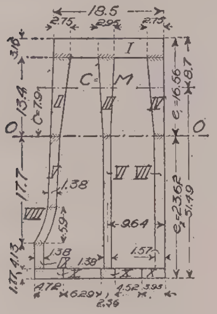


FIG. 94.—Dimensions in Inches.

$$P=\frac{.785\times 35.4^2\times 450}{2}=221\,000 \text{ lbs.}$$

The area of section is 260 sq.in., so that the purely tensile stress is

$$\sigma=\frac{221\,000}{260}=850 \text{ lbs. per sq.in.}$$

Distance between neutral axis and cylinder axis is $c=7.9''$, so that the maximum bending moment is

$$M_b=7.9\times 221\,000=1\,740\,000 \text{ in.-lbs.}$$

Moments of inertia of the partial areas (the distances of the center of gravity of each area from the neutral axis of the section are not given in the figure):

$$\left.\begin{aligned}
I_{\text{I}}&=\frac{18.5\times 3.15^3}{12}+(18.5\times 3.15\times 14.9^2)=12\,998 \text{ in.}^4\\
I_{\text{II}}&=\frac{2.06\times 13.4^3}{12}+(2.06\times 13.4\times 7.3^2)\ =\ 1\,883 \text{ in.}^4\\
I_{\text{III}}&=\frac{2.16\times 13.4^3}{12}+(2.16\times 13.4\times 7.3^2)\ =\ 1\,968 \text{ in.}^4\\
I_{\text{IV}}&=\frac{2.16\times 13.4^3}{12}+(2.16\times 13.4\times 7.1^2)\ =\ 1\,893 \text{ in.}^4\\
I_{\text{V}}&=\frac{1.38\times 11.8^3}{12}+(1.38\times 11.8\times 5.9^2)\ =\ 754 \text{ in.}^4\\
I_{\text{VI}}&=\frac{1.38\times 21.8^3}{12}+(1.38\times 21.8\times 10.8^2)=\ 4\,690 \text{ in.}^4\\
I_{\text{VII}}&=\frac{1.57\times 21.8^3}{12}+(1.57\times 21.8\times 10.8^2)=\ 5\,390 \text{ in.}^4\\
I_{\text{VIII}}&=\frac{1.38\times 5.9^3}{12}+(1.38\times 5.9\times 14.8^2)\ =\ 1\,808 \text{ in.}^4\\
I_{\text{IX}}&=\frac{1.38\times 4.13^3}{12}+(1.38\times 4.13\times 19.8^2)=\ 2\,248 \text{ in.}^4\\
I_{\text{X}}&=\frac{11\times 1.77^3}{12}+(11\times 1.77\times 22.7^2)\ =10\,005 \text{ in.}^4\\
&\qquad\qquad\qquad\qquad\qquad\qquad\qquad\qquad =43\,637 \text{ in}^4.
\end{aligned}\right\}\ \begin{gathered}\text{Moment of inertia of}\\ \text{section of side wall}\\ \Sigma(I_n)=45\,614 \text{ in.}^4\end{gathered}$$

Distance of upper fiber from neutral axis of section is $13.4+3.16=16.56''$. Hence section modulus is

$$W=\frac{43\,637}{16.56}=2630,$$

and maximum bending stress will be

$$\sigma_b = \frac{1\,740\,000}{2630} = 660 \text{ lbs. per sq.in.}$$

Total fiber stress is therefore

$$\sigma + \sigma_b = 850 + 660 = 1510 \text{ lbs. per sq.in.}$$

From this example the good effect of proper design of cross-section upon the fiber stress is clearly seen. In spite of the large cylinder diameter, the distance from the center of the cylinder to the neutral axis of the section is only 7.9″, and the bending stress is in this case not larger than the tensile stress, as is commonly the case, but is about one fourth smaller.

4. In the Nürnberg blast-furnace gas engine, shown in Plate XVIII, cylinder diameter 52″, the dangerous section of the frame is that shown in Fig. 95. Each beam constituting the side walls is cast in two pieces, and the soft steel tie-rod XIII, Fig. 95, placed in the neutral axis of the section serves, in the first place, to unite the parts of the side wall, and in the second place relieves the castings of some of the stress.

Assuming the explosion pressure to be 430 lbs., the maximum load on each beam of the frame will be

$$P = \frac{430 \times .785 \times 52^2}{2} = 455\,000 \text{ lbs.}$$

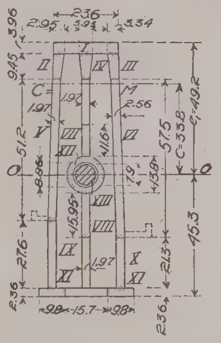


FIG. 95.—Dimensions in Inches.

It is not known how this load is distributed between the casting and the tie-rod; it is therefore necessary to make some kind of an estimate. Assume that the tie-rod is subjected by the load P to a stress equal to 4250 lbs. per sq.in. In that case the purely tensile stress on the cast-iron section, whose area is about 776 sq.in., will be

$$\sigma = \frac{455\,000 - (.785 \times 7.9^2 \times 4250)}{776 - 48.3} = 346 \text{ lbs. per sq.in.}$$

Moments of inertia of the partial areas referred to the neutral axis of the entire section:

$$I_{I} = \frac{23.6 \times 3.96^3}{12} + (23.6 \times 3.96 \times 47.3^2) = 209\,120 \text{ in.}^4$$

$$I_{II} = \frac{2.46 \times 9.45^3}{12} + (2.46 \times 9.45 \times 40.6^2) = 38\,370 \text{ in.}^4$$

$$I_{III} = \frac{2.96 \times 9.45^3}{12} + (2.96 \times 9.45 \times 40.6^2) = 46\,300 \text{ in.}^4$$

$$I_{IV} = \frac{2.96 \times 9.45^3}{12} + (2.96 \times 9.45 \times 40.6^2) = 46\,300 \text{ in.}^4$$

$$I_{V} = \frac{1.97 \times 51.2^3}{12} + (1.97 \times 51.2 \times 10.05^2) = 32\,200 \text{ in.}^4$$

$$I_{V} = \frac{2.56 \times 57.5^3}{12} + (2.56 \times 57.5 \times 7.1^2) = 48\,030 \text{ in.}^4$$

$$I_{\text{VII}}=\frac{1.97\times 11.6^3}{12}+(1.97\times 11.6\times 12.6^2)\;=\;3880\text{ in.}^4$$

$$I_{\text{VIII}}=\frac{1.97\times 15.95^3}{12}+(1.97\times 15.95\times 14.8^2)=\;7540\text{ in.}^4$$

$$I_{\text{IX}}=\frac{1.97\times 27.6^3}{12}+(1.97\times 27.6\times 30.3^2)\;=\;53440\text{ in.}^4$$

$$I_{\text{X}}=\frac{2.56\times 21.3^3}{12}+(2.56\times 21.3\times 32.3^2)\;=\;58760\text{ in.}^4$$

$$I_{\text{XI}}=\frac{19.6\times 2.36^3}{12}+(19.6\times 2.36\times 44.1^2)\;=\;90020\text{ in.}^4$$

$$I_{\text{XII}}=.05(13.8^4-8.86^4)\;=\;1500\text{ in.}^4$$

$$I_{\text{XIII}}=\frac{\pi\times 7.9^4}{64}\;=\;190\text{ in.}^4$$

Moment of inertia of entire cross-section $=635650$ in.^{4}

including the tie-rod.

After subtracting partial moment I_{XIII} there remains for the cast-iron cross-section

$$I=635\,650-190=635\,460\text{ in.}^4$$

Hence the section modulus, with $e_1=49.2''$

$$=W=\frac{I}{e_1}=\frac{635460}{49.2}=12920.$$

The section modulus of the tie-rod is

$$\frac{190}{3.94}=48.2.$$

The maximum bending moment on the entire section, with $c=33.8''$, is

$$M_b=455\,000\times 33.8=15\,400000\text{ in.-lbs.}$$

Since it is assumed that the tie-rod is put under a stress of 4250 lbs. per sq.in., by the load P, the part of the above moment M_b falling to the rod will be

$$4250\times 48.2=205\,000\text{ in.-lbs.}$$

The maximum bending moment for the cast section alone is therefore

$$M_b=15\,400000-205\,000=15\,195\,000\text{ in.-lbs.}$$

and hence the bending stress in the outer fiber is

$$\sigma_b=\frac{15\,195\,000}{12\,920}=1175\text{ lbs. per sq.in.}$$

The maximum total stress on the outer fiber of the cast iron is therefore

$$\sigma+\sigma_b=346+1179=1525 \text{ lbs. per sq.in.}$$

The maximum stress in the outer fiber of the steel tie-rod will be

$$4250+4250=8500 \text{ lbs. per sq.in.}$$

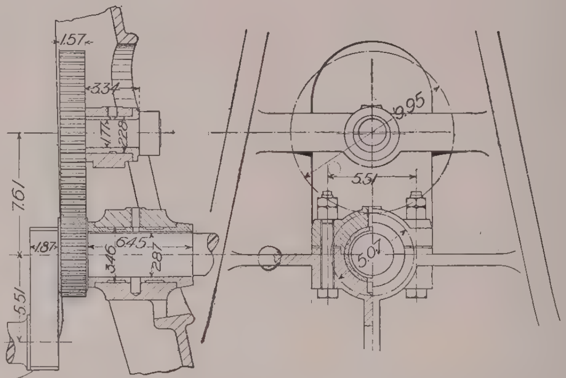
To this must be added the constant but unknown stress induced in the rod by the drawing up of the nuts. This stress, however, is of comparatively small importance.

In case the cylinder is overhung, the connecting flange in the frame must be made abundantly wide and strong, because this flange, besides being loaded by the explosion pressure P_z is also subject, during the expansion stroke, to the lateral thrust N of the connecting rod, which may under certain circumstances cause a considerable bending moment in the flange. The maximum value of N is approximately $\frac{P_z}{10}$, and this maximum occurs at from 15–18% of the stroke. Consequently, the point of application of $N_{\max}$ is much nearer to the head of the cylinder than to the connecting flange, especially if the piston pin is well toward the face of the piston, and the moment arm on the flange is correspondingly long. For this reason the overhung cylinder should be used in small engines only. If, however, the cylinder proper is merely a liner inserted into a barrel extension of the frame casting, the barrel forming the jacket wall of the cylinder, this jacket wall must take care of the load due to P_z, which is always considerably in excess of the bending stress due to $N_{\max}$. The latter is taken care of by both the jacket wall and the liner. For further details on these points, see "Cylinders," p. 129.

6. The best form for the **main bearings** for horizontal engines is the one in which the explosion load is taken up directly by the frame. This is possible only when the dividing line of the bearing slants inward, Figs. 71–80, or is horizontal. Bearings slanting outward, Fig. 107, are not suitable because the maximum load comes on the caps and bolts. Further, the journals run upon the lower dividing line of the bearing all the time, because the influence of the weight of the fly wheel upon the direction of pressure on the journal is of small importance. In view of these disadvantages, the good points that this type of bearing may have are decidedly overbalanced. The most suitable form of main bearing is undoubtedly that slanting inward, for the explosion load is always taken up by the frame body directly and only the lower half of the bearing is ever under load. But this form requires low frame walls, and since these are not suitable for the larger models, the proper use of this type of main bearing is necessarily confined to the smaller machines. For the large horizontal gas engine there remains only the horizontally divided main bearing.

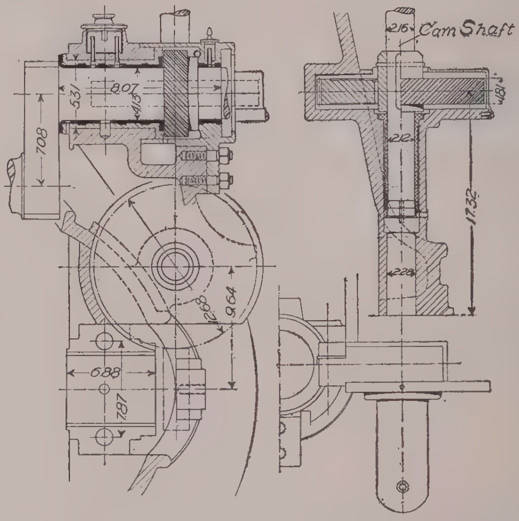
In vertical machines, with crank shaft below the cylinder, the journal pressure due to P_z always falls to the lower half of the bearing. It should be noted, however, that the weight of the fly-wheel in this type of engine causes an upward pressure on the cap of the opposite bearing when the piston is unloaded. This pressure is, however, never great enough to determine the dimensions of either the cap or the bolts. The last statement also holds for vertical frames with shaft above the cylinder. In this type of frame the direction of the journal or piston pressure is always through the cap of the bearing, whose dimensions are to be determined accordingly.

The lubrication of main bearings should be thorough, certain, and easy of inspection. Drop-feed oilers are suitable for small machines only. For medium-sized engines ring oiling is much in favor, although this method also has many weak points (Certain cutting down and weakening of the bearing surfaces, operation not directly open to inspection, gradual deterioration of the lubricant, etc.) For the bearings of large engines, forced lubrication by means of special oil pump is the best method.

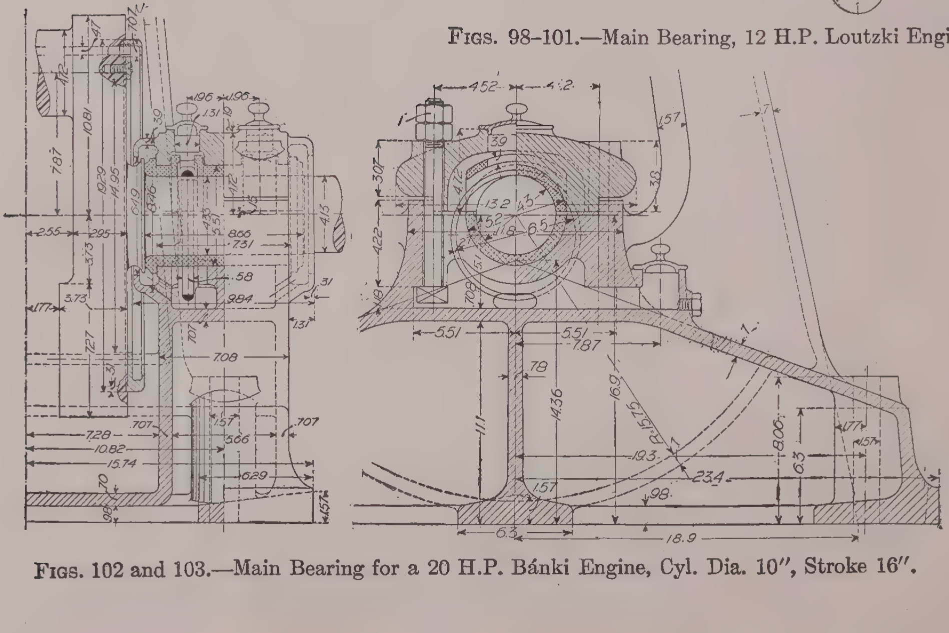


Figs. 96 and 97.—Main Bearing, small Loutzki Engine.

Designs of Main Bearings.



Figs. 98–101.—Main Bearing, 12 H.P. Loutzki Engine.

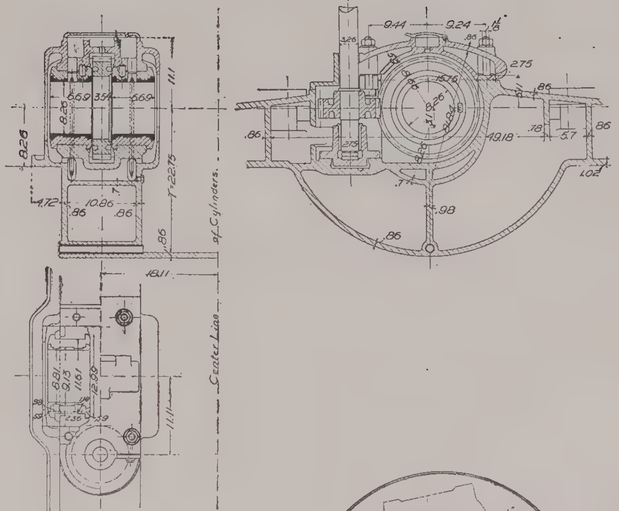


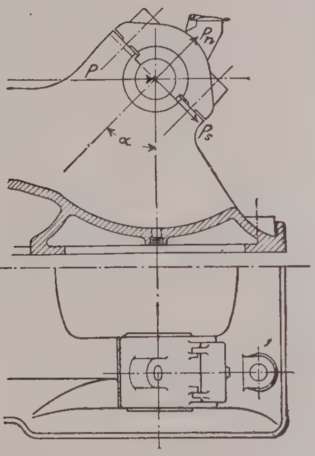
Figs. 102 and 103.—Main Bearing for a 20 H.P. Bánki Engine, Cyl. Dia. 10″, Stroke 16″.

FIGS. 104–106.—Main Bearing for an 80 H.P. Vertical Diesel Engine, dia. 16″, stroke 24″. Built by Maschinenfabrik, Augsburg.

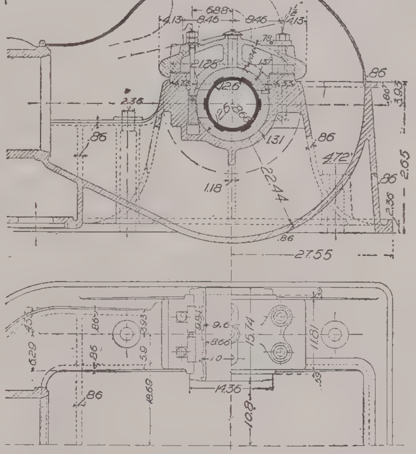
The gears driving the intermediate shaft are located in the center-plane of the left hand main bearing.

(This construction was originally used by Loutzki.)





FIGS. 107 and 108.—Type of Bearing used on Körting Engines of from 2–16 H.P.



FIGS. 109 and 110.—Bearing for Frame shown in Figs. 81 and 82.

Constructive Details. For diameter and length of bearing, or allowable bearing pressures, see Crank-shafts, p. 160.

The material for bearings up to 5″ journal diameter (d) is nearly always bronze, in which case the thickness of the metal may be made

$$s=\left(\frac{d}{15}+.20\right) \text{ inches.} \qquad (4)$$

For larger journal diameters the bearings are made of cast iron, lined with white metal, Figs. 111 and 112. The least thickness of the shell of the bearing proper (without the lining) may be made about

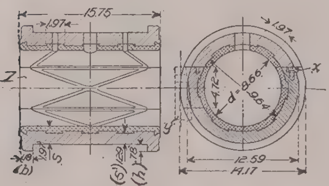
$$s'=\left(\frac{d}{10}+.4\right) \text{ inches.} \qquad (5)$$

In either case the collars at the end of the bearing may be given, see Figs. 111 and 112,

$$\left.\begin{array}{ll}\text{a radial height,} & h=\ .8 \text{ to } 1.1\ s \text{ or } s', \\ \text{and an axial thickness,} & b=1.0 \text{ to } 1.2\ s \text{ or } s'.\end{array}\right\} \quad . \ . \ . \ . \ . \ . \ . \ . \ . \ . \quad (6)$$

If special circumstances make it necessary to keep the external diameter of the bearing as small as possible, cast steel or malleable iron may be used in place of cast iron. In any case a white metal liner is employed. The thickness of the walls s may then be taken according to eq. (4), the same as for bronze.

On account of sudden variations of pressure, the proper choice of the white metal alloy[1] and the manner of holding it in place are of the greatest importance. The shell of the bearing should be furnished with sufficient dove-tail grooves both circumferentially and longitudinally, Figs. 111 and 112, and the surfaces should be well tinned over before the lining is cast in. If these precautions are not taken, the lining is apt to be squeezed out of shape and loosened in the shell. Care should also be taken that the ends of the bearings, where they may come in contact with the crank shaft, taking up any lateral pressure that may occur, are well covered with white metal.



Figs. 111 and 112.

For reasons of economy in manufacture, the outside surface of the bearings is without any projection, so that it can be finished in the lathe. This permits the bearing to rest evenly over its entire length in its supports. The shifting or turning of the bearing in its seat is prevented by safety pins or plates afterward inserted or screwed on (see x, Fig. 112). The lower half of the bearing is not thus secured in order to be able to take it out, in case of necessity, by slightly raising the shaft and turning the half bearing about the shaft until it comes on top.

In the larger horizontal machines lateral adjustment in the bearings may have to be provided for. In such a case the adjusting mechanism should always be placed in the side of the bearing nearest the cylinder in order to protect the adjusting screws or wedges against the full load on the bearing. Three part bearings, Figs. 111-112, may in such cases be successfully employed. The adjusting wedge, y, and the movable block, z, are placed in the side of the bearing under least stress, so that all joints are out of the direct line of journal pressure.

Examples. 1. The main bearing, Fig. 107, slanting outwards at an angle $\alpha=40°$ with the vertical, is to carry a load of $P=\frac{1}{2}P_2=9900$ lbs. The component P_n of the force P, passing normally through the cap of the bearing will be

$$P_n=P \sin \alpha=9900 \sin 40°=9900\times .643=6380 \text{ lbs.}$$

This is the load which determines the dimensions of the caps and bolts by the method of Example 1, p. 89. The load on the bearing furthest from the fly-wheel is relieved somewhat by the weight of the wheel, but an additional load is often caused by the pull of the driving belt, and hence this should not be taken into account. Pre-ignition also causes an increase in P_n, and for that reason, too, a good factor of safety is essential.

---

[1] Good alloy for heavily loaded main bearings: 8 parts by weight of copper, 12 of antimony, and 80 of tin.

2. The main bearing shown in Fig. 113 slants inward. The load P at the moment of explosion is 11 000 lbs. This horizontal force causes a shearing stress in the section x–x whose area is 11 sq.in., of

$$T=\frac{11\,000}{11}=1000 \text{ lbs. per sq.in.}$$

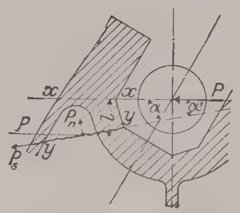


FIG 113

Fracture, however, would in all probability not take place in the section x–x, but in y–y. This dangerous section must therefore also be examined. Any influence that the cap may have to relieve the load on this section should always be neglected. The component P_n, acting at the neutral axis of the section y–y is equal to $P_n=P\sin\alpha'=11\,000\times\sin 20°=3760$ lbs. The area of the section, Fig. 114, is 12.6 sq.in., hence the tensile stress is

$$\sigma=\frac{3760}{12.6}=297 \text{ lbs. per sq.in.}$$

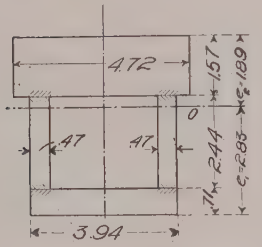


FIG. 114.—Dimensions in Inches.

Besides this there is acting on this section a bending moment

$$M_b=Pl=11\,000\times 1.65''=18\,100 \text{ in.-lbs.}$$

The cross-section y–y is shown in Fig. 114. From this the moment of inertia

$$I=I_{\text{I}}+I_{\text{II}}+I_{\text{III}}+I_{\text{IV}}=\left[\frac{4.72\times 1.57^3}{12}+(4.72\times 1.57\times 1.1^2)\right]$$

$$+2\left[\frac{.47\times 2.44^3}{12}+(.47\times 2.44\times .71^2)\right]$$

$$+\left[\frac{3.94\times .71^3}{12}+(3.94\times .71\times 2.48^2)\right]=30.2 \text{ in.}^4$$

The maximum bending stress is therefore

$$\sigma_b=\frac{M_b e_2}{I}=\frac{18\,100\times 1.89}{30.2}=1130 \text{ lbs. per sq.in.}$$

The combined stress, $\sigma+\sigma_b=297+1130=1427$ lbs. per sq.in., would be much smaller if the bearing had a cylindrical seat.

3. The following is a systematic investigation made by Professor Bach on a fractured main bearing of a steam engine.[1] It serves to show very clearly to what extent the safety of a construction sometimes depends upon the judgment of the designer, and how, in the case of main bearings, the usual methods of computation are sometimes badly misleading.

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[1] Zeitschrift d. V. D. I., 1901, p. 1567.

Fig. 115 shows the fractured pedestal, one break, *a–b*, starting from the corner of the seat of the bearing and extending diagonally toward the base, the other, *c–d*, is in the cap. To obtain some idea of the load at rupture by actual measurement, cast-iron test specimens, Fig. 116, were made in which the dangerous section, under stress due to the force P, had dimensions as close as possible to those of the actual surface of the break in the pedestal. These specimens broke, through the section indicated in Fig. 116, under a load $P=46700$ lbs. The dimensions of the cross-section are given in Fig. 117.

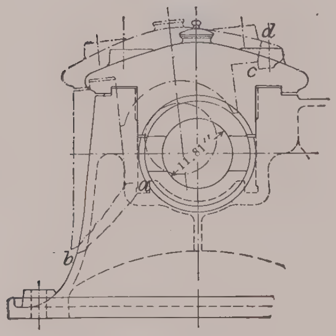

FIG. 115.

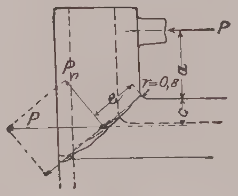

FIG. 116.—Dimensions in Inches.

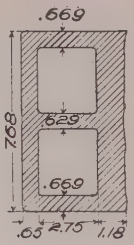

FIG. 117.—Dimensions in Inches.

The common method of computation, using the dimensions and data given in the figures, then gives the following results:

$$P=46700 \text{ lbs.}, \qquad \alpha=40°, \qquad P_n=46700 \sin 40°=29000 \text{ lbs.}, \qquad a=2.92 \text{ in.}, \qquad c=1.23 \text{ in.},$$

$$e=\frac{(7.68\times 4.58\times 2.29)-(5.72\times 2.75\times 2.56)}{19.45}=2.07 \text{ in.},$$

in which $19.45=f=$area of the cross-section. Moment of inertia of the section of the break is

$$I=\left[\frac{7.68\times 4.58^3}{12}+7.68\times 4.58(2.29-2.07)^2\right]$$

$$-\left[\frac{5.72\times 2.75^3}{12}+5.72\times 2.75(2.56-2.07)^2\right]=50 \text{ in.}^4$$

The section modulus is therefore

$$W=\frac{I}{e}=\frac{50}{2.07}=24.2.$$

Finally, the stresses involved are

$$\left.\begin{aligned} &\text{from } P_n, && \sigma=\frac{P_n}{f}=\frac{29000}{19.45}=1490 \text{ lbs. per sq.in.,} \\ &\text{and from } P, && \sigma_b=\frac{P(a+c)}{W}=\frac{46700\times 4.15}{24.2}=8000 \text{ lbs. per sq.in.} \end{aligned}\right\} \quad \ldots\ldots\ldots \quad (7)$$

Test specimens of the same cast iron under transverse tests showed a ratio of strength in the outer fiber of $K_b:K_z$[1]$=1.79:1$, and a strength in the outer fiber of $K_b=33300$ lbs. per sq.in.

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[1] See Bach, Elasticität and Festigkeit, 2 ed., eq. (77) or Maschinen Elemente, 8 ed., eq. (137).

The above computation, based on actual tests, however, shows a total stress in the fractured section of only

$$1.79\sigma+\sigma_b=(1.79\times1490)+8000=10700 \text{ lbs. per sq.in.},$$

which is $1-\frac{10700}{33300}$ = two-thirds smaller than the actual strength of the material in the outer fiber.

The above method of computation therefore seriously underestimates the actual existing stress, and the reason is that the outside fiber, which is under greatest stress has been considered as straight, while in reality, owing to the fillet put in with a radius equal to .31″, it must be considered as an element with considerable curvature. The moment due to P tends to straighten out this fiber, and of course intensifies the stress very materially, as may be shown as follows.

Since the radius of the curvature of the fillet is .31″, the mean radius of curvature for the neutral plane of the section will be $r=.31+e=.31+2.07=2.38''$. The stress induced in curved rods, according to Bach,[1] is

$$\sigma=\left[\frac{P_n}{f}+\frac{M_b}{fr}+\frac{M_b}{xfr}\times\frac{e}{r+e}\right] \text{ lbs. per sq.in.}, \quad \ldots\ldots\ldots\ldots \quad (8)$$

in which, from the above figures, $P_n=29000$ lbs., $f=19.45$ sq.in., $M_b=-46700\times(2.92\times1.23)$ $=-193500$ in.-lbs.,[2] and $x=\frac{1}{f}\int\frac{e}{r+e}df=.844$.[3]

The maximum stress in the outer fiber will then be, with $e=-2.07''$,

$$\sigma_{\text{max}}=\frac{29000}{19.45}+\frac{-193500}{19.45\times2.38}+\frac{-193500}{.844\times19.45\times2.38}\times\frac{-2.07}{2.38-2.07}$$

$$=1490-4180+33100=30410 \text{ lbs. per sq.in.}$$

As above mentioned, transverse tests made on rectangular rods cut from the specimens of Fig. 116, gave a tensile stress in the outer fiber of 33300 lbs. per sq.in. To derive from this the strength of the material under pure tension, we may use the following formula due to Bach.[4]

$$K_b=K_z\mu_0\sqrt{\frac{e_1}{z_0}}, \quad \ldots\ldots\ldots\ldots\ldots \quad (9)$$

in which K_b = tensile strength per sq.in. in outer fiber;
K_z = tensile strength, pure tension;
μ_0 = a factor depending upon shape of the cross-section and whether the surfaces retain the cast skin or not. In this case the skin is retained and μ_0 may be taken = 1.0;
e_1 = maximum fiber distance;
z_0 = distance from neutral axis of the entire section to the neutral axis of that part of the cross-section on one side of the first mentioned axis and toward the fiber under greatest stress.

For a rectangular section $z_0=\frac{1}{2}e_1$, so that we may write for the tensile stress of the rods which gave $K_b=33300$ lbs. under transverse stress

$$K_b=33300=K_z\sqrt{\frac{e_1}{.5e_1}}=1.414K_z,$$

from which

$$K_z=\frac{33300}{1.414}=23500 \text{ lbs. per sq.in.}$$

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[1] Elasticität and Festigkeit, eq. (8) or Maschinen Elemente, eq. (103).

[2] The minus sign is used because the moment tends to decrease the curvature.

[3] For graphical solution, see Zeitschrift d. V. D. I. 1901, p. 164.

[4] Bach, Elasticität and Festigkeit, art. 22, eq. (1), or Maschinen Elemente, 8 ed., eq. (137), p. 43.

Now to refer this to a cross-section like that of Fig. 117, we have $e_1=2.07''$, $z_0=1.32''$, and $K_z=23500$, hence for this section

$$K_b=23500\sqrt{\frac{2.07}{1.32}}=23500\times 1.26=29700 \text{ lbs. per sq.in.}$$

This figure agrees very well with 30400 lbs. per sq.in., above determined by computation, and this would show that eq. (8) is the most reliable for similar cases.

It should be noted that the stress will be the smaller the greater the value of r, the radius of the fillet. The radius of curvature should, therefore, be made as large as possible, which finally leads to the use of cylindrical seats for the main bearings.

If the much simpler equations (7), which apply to straight prismatic rods, are used for approximate computations, the allowable fiber stress should be assumed only one-third as great as would ordinarily be considered the limit. If the stress in the dangerous section is not to exceed, say, 4200 lbs. per sq.in., the dimensions must be chosen such that equations (7) do not show a total stress exceeding 1400 lbs. per sq.in. It is evident, however, that in important cases an accurate determination of the stress conditions, according to eq. (8), is absolutely necessary.

The bolts or studs connecting the cylinder to the frame must be computed with P_z as the maximum total load. With reference to anchor bolts, see pp. 90 and 93.

II. Cylinders and Jackets

1. Cylinders. *Material:* Hard, close-grained cast iron. Allowable stress fairly high to reduce the thickness of walls with reference to expansion by heat. Sufficient stiffness should be maintained. For cast iron with fairly clearly defined stress conditions, the allowable fiber stress may be $K_b=4250$ lbs. per sq.in., in other cases less.

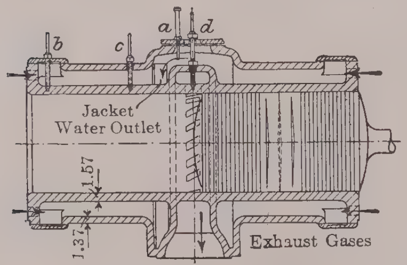


FIG. 118.—Cylinder of a Körting 2-cycle Engine.

Cylinders and jackets, and sometimes frames, of small machines are very often cast in one piece to cheapen the cost of construction; the larger sizes, commencing, say, with 8″ diameter of cylinder, are mostly made with a separate cylinder liner, if special conditions, as for instance radial inlet and outlet parts, etc., do not compel the use of the one-piece construction. A separate cylinder liner or barrel has the advantages that a specially suitable grade of iron can be employed, that the jacket and frame are cast with greater ease, and that the wear on the cylinder can be taken care of more readily. To this should be added the special advantage, important in gas-engine construction, that the separate liner can expand axially independent of the jacket, which is very important on account of the great temperature difference in these two parts. To allow of this free expansion, the liner is to be rigidly fastened only at its inner end, the other end sliding axially in its seat. Of course, the one-piece cylinder also has its advantages, in that it is a stiffer construction than the other, and for that reason its use would seem to recommend itself especially for large engines; but greater difficulty of manufacture and repair, unequal expansion under heat, etc., counterbalance this gain.

The temperature differences existing between the inner and outer walls of the cylinder and their influence upon the expansion or stress conditions in these important parts have been investigated but little. They are, however, certainly greater than is commonly assumed. Of all the direct temperature measurements undertaken, those made by Ernest Körting, in 1901, on one of his large double-acting 2-cycle machines are of greatest value to the designer. The cylinder of a 400 H.P. engine of this type was fitted with thermometers *a–d*, as indicated in Fig. 118. The bulbs of these thermometers were placed exactly at the center of the inner wall. Tubes passing through the jacket space and having their lower ends filled with mercury, were inserted as shown, so that the centers of the thermometer bulbs were $\frac{7}{8}''$ from the bore of the cylinder.

Table 9

TEMPERATURES OF THE INNER CYLINDER WALL OF KÖRTING ENGINES

Hour		3:30	4:00	4:15	4:30	4:45	5:00	5:15
Load on engine		$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$	full
Temperature of cooling water, outlet ° F		91	95	97	99	99	99	101
Temperature of cylinder wall at point	*b*, ° F.	167	167	181	194	190	198	201
	c, ° F.	134	134	147	150	143	145	145
	d, ° F.	314	316	321	330	338	338	338

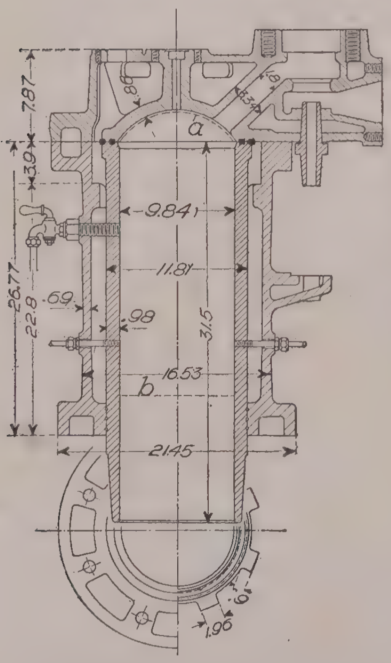
The measurements were made with different loads on the engine. From Table 9 it is seen how quickly the temperature changes when the load on the engine varies, i.e., when the quantity of heat developed changes. It will also be seen that the temperature of the cylinder wall near the combustion chamber is considerably higher than at a point about mid-stroke, in spite of the fact that the point *b*, on account of its position near the dead center, is covered by the water-cooled piston, at least for a time equivalent to one-quarter of the stroke. The conclusion from this is that the temperature of the combustion chamber, which is subject to much higher gas temperatures, and which is at no time cooled from the inside, must be much higher than the temperatures at *b*. The highest temperature determined at full load is 338° It is unquestionable that the temperature of the inner wall is higher than this, especially if the piston be not water cooled, or shorter than in this type of machine. If the temperature of the jacket wall is assumed equal to the mean temperature of the jacket water, i.e. $=84°$ F., and the mean temperature of the cylinder wall equal to $\frac{201+145}{2}=173°$, neglecting the temperature near the exhaust ports, the expansion of the cylinder, relative to that of the jacket wall will be $=(173-84)\times .00000593=.000528$. Even if this entire expansion were to be taken up by the jacket wall alone, the stress induced would still be below the elastic limit of cast iron, the extension of which in tension is on an average not below .00075 at the elastic limit. The case, however, becomes a little more serious when the mean wall temperature t_m, including the exhaust port temperatures is used in the computations. t_m will be $=(201+145+338)\div 3=228°$ F., and the expansion relative to the jacket wall will be $=(228-84)\times .00000593=.000854$. This already exceeds the average extension of cast iron under tension by 13%. Now it is, of course, a fact, that in one-piece cylinders the axial stresses are not carried by the cylinder wall alone, but by the cylinder and jacket walls combined. This, under the most favorable circumstances, reduces the stresses by one-half. On the other hand, the elastic limit strength of cast iron is considerably reduced by repeated heating and cooling, such as exists in gas engines, i.e., the iron grows more brittle, and is more liable to fracture. In general, too, the outlet temperature of the jacket water is much higher than that used in the above tests, and the internal cooling of the cylinder wall is much less effective with pistons not water cooled. Finally, the stresses to which the cylinder is subjected by the explosion pressure in both radial and axial directions enter the problem. Under such conditions it may happen—and practical experience proves it—that the sum total of the stresses due to expansion by heat, and to the working load on the cylinder may far exceed the elastic limit strength of the material, leading to serious damage by fracture or warping, if the cylinder is cast in one piece.

It is not easy to relieve the jacket walls of double-acting cylinders, with valve cages cast in one piece with the cylinder, of longitudinal temperature stresses. The use of a separate cylinder barrel is in the case of such cylinders no longer practicable, and it becomes necessary to divide the jacket wall into two parts, longitudinally, so that the two half cylinders so formed, surround the cylinder wall proper in saddle shape. The construction is shown in the double-acting Deutz engine, Plates IX and X. Langen and Wolf, in their new machines designed by Ebbs, use the method of relieving the jacket wall described in Part III under III, 1, in which case the one-piece construction may be retained.

Injurious temperature stresses may also be caused by injudicious distribution of metal, for which reason it is well, in the case of large cylinders, to design with an eye to symmetry, at least, as far as the combustion chamber and the adjacent parts are concerned. Any violation of this rule also introduces trouble owing to the fact that unequal distribution of metal causes irregular radial expansion of the cylinder bore, which of course, leads to cutting and leaking between cylinder wall and piston.

Designs of Cylinders.

The water jacket should, if possible, cover the entire length of the stroke, even in single-acting machines. If the open end of the cylinder is left unjacketed, there will be unequal expansion in the cylinder bore, and annoying vapors due to burning lubricating oil are apt to develop. All water spaces should be of ample width, and in the case of large cylinders sufficient cleaning openings should be provided to rid the water space of core sand in the course of manufacture, and of any sediment or scale that may form during operation. Naturally, the cold water should be admitted to the jacket at the hottest part of the cylinder, and after this it should be so conducted, by means of baffles, if necessary, that nowhere in the jacket space shall there be a chance of impeded circulation or formation of air pockets.



Figs. 119 and 120.—Cylinder and Cylinder Head for a 20 H.P. Bánki Engine. (The broken lines *a* and *b* indicate the upper and lower ends of the piston in its highest position.)

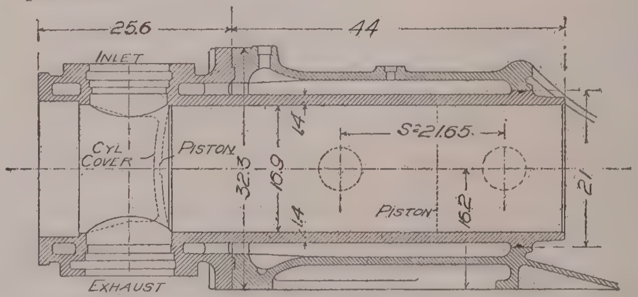
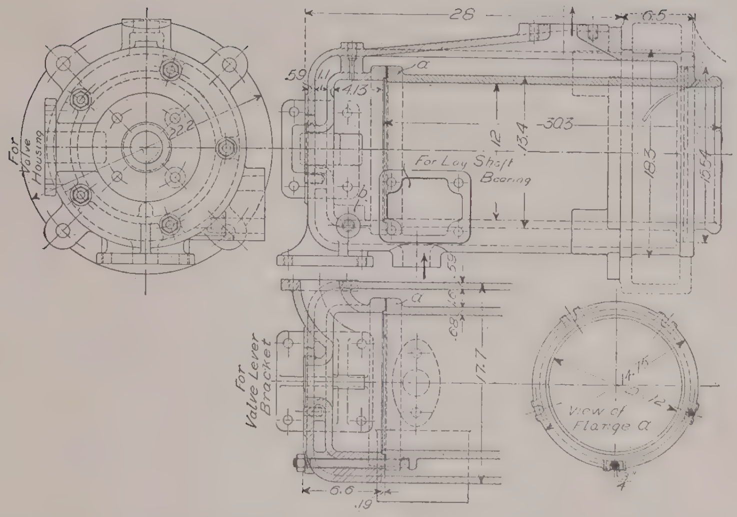


Fig. 121.—Cylinder for a 60 H.P. Engine, Langen & Wolf, Vienna. (Cylinder head in one piece with liner.)



Figs. 122–125.—Cylinder for a 12–15 H P. Hornsby-Akroyd Engine of German Make. (Projection *b* is to prevent the piston from going too far back when assembling.)

Constructive Details. The volume of the combustion chamber may be computed from eq. (55) p. 32. From the compression ratio ε, required for the given compression pressure p_c, the length h of the combustion chamber, when S is the length of the stroke, will be

$$h=\frac{S}{\varepsilon-1}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (10)$$

assuming the combustion chamber to have the same cross-section as the cylinder bore.

The explosion pressure p_z, which determines the cylinder dimensions, causes tensile stresses in the cylinder walls, both in a longitudinal and a radial direction. To these should be added, under circumstances, secondary stresses due to drawing up of the flange nuts, the thrust of the connecting rod, unequal temperature stresses, etc. In the case of one-piece cylinders, the tensile stress in the longitudinal direction may be neglected, if the inner cylinder wall is properly designed to withstand the radial stress due to p_z, because this latter stress is always more than twice the former. If, on the other hand, jacket wall and cylinder wall are separate, the jacket wall alone is usually called upon to carry the longitudinal stress or the stress in the axial direction, and since the jacket wall is usually computed on a basis of load much less than that due to p_z, care should be taken in this construction to see that the axial stress does not become dangerous. This is especially important for places in the jacket wall that are expected to carry an additional external load or that have been weakened in any way.

According to Bach,[1] if we let

R_a = external radius of cylinder in inches,
R_i = internal radius of cylinder in inches,
K_z = allowable tensile stress for the material,
and p_i = internal pressure pounds per sq.in.,

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[1] Bach, Maschinen Elemente, 8 ed., p. 36.

we shall have the relation

$$R_a = R_i \sqrt{\frac{1+.4\dfrac{p_i}{K_z}}{1-1.3\dfrac{p_i}{K_z}}} = R_i \sqrt{\frac{K_z+.4p_i}{K_z-1.3p_i}} \text{ inches.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (11)$$

From this, if s is the thickness of the cylinder wall, D = diameter of cylinder in inches, and $p_i = p_z$.

$$s = \frac{D}{2}\left(\sqrt{\frac{K_z+.4p_i}{K_z-1.3p_i}}-1\right) \text{ inches.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (12)$$

Since p_z acts only at the inner end of the cylinder barrel which is nearly always reinforced by a flange, the value of K_z may be taken correspondingly high, from 3200–3500 lbs. per sq.in. for cast iron.

The normal indicator diagram then calls for a thickness of inner cylinder wall of

$$s = \frac{D}{2}\left(\sqrt{\frac{3500+140}{3500-460}}-1\right) = \frac{D}{2}(\sqrt{1.197}-1) = .047\,D \text{ inches.} \quad . \quad . \quad . \quad . \quad (12n)$$

To this should be added, as an allowance for reboring, from $\frac{1}{4}$ to $\frac{1}{2}$ in. For practical reasons (possible shifting of cores, etc.), it is also necessary that, in the case of stationary machines, up to say 8″ diameter, s should be at least $\geqq .7''$, for one-piece cylinders, and at least $\geqq .65''$ for separate cylinder barrels.

In cylinders over 25″ diameter, the thickness of the wall may, for reasons of economy and decrease in weight, be decreased gradually or by steps until at the open end

$$s_0 = .02\,D + .5 \text{ in.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (12a)$$

provided additional stresses or other circumstances do not forbid this.

Due to the pressure p_z, the inner fibers of the wall are under a greater stress than the outer fibers. This inequality of stress increases materially with an increase in the thickness of the wall. For that reason s should not be made thicker than necessary. Excessive thickness of cylinder wall is also wrong from the standpoint that the cooling water is less effective, consequently the conduction of heat is less rapid, and troublesome overheating of the material may ensue.

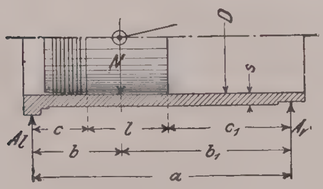

FIG. 126.

In large or long-stroke engines, the thrust of the connecting rod should receive careful attention in dimensioning the material of the cylinder. In case the dimensions of the inner wall, as determined by eq. (12) should furnish a section modulus too small to take care of the normal pressure N due to the connecting rod, it is better to increase the stiffness of the cylinder wall by suitable supports or webbing than to increase the thickness s.

With the favorable assumption that N is distributed over the bearing surface of length l, Fig. 126, the maximum bending moment will be

$$M_b = A_l\left(c+\frac{lA_l}{2N}\right) \overset{=}{<} W K_b. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (13)$$

In eq. (13) the maximum pressure at the supports is

$$A_l = N\frac{2c_1+l}{2a} \text{ pounds.} \tag{14}$$

In case a separate cross-head is used, l is the length of the shoe, with trunk pistons, l is the length of the piston body in actual contact, i.e., excluding any part made smaller to give play for expansion.

The stiffness of the cylinder liner, or the cylinder, should be such that N cannot cause any noticeable bending or deformation. In order to be sure of this point it is advisable also to make a check computation, assuming that the force N, instead of being distributed over the length l, is concentrated at the middle of this length. In that case

$$M_b = N\frac{bb_1}{a} \text{ in.-lbs.} \tag{15}$$

and assuming that K_b is the allowable stress, the section modulus required will be

$$W = N\frac{bb_1}{aK_b} \geqq \frac{M_b}{K_b}. \tag{16}$$

Assuming that the moment of inertia of the cross-section of the cylinder is I and that the coefficient of extension is α, (α = reciprocal of modulus of elasticity), the deflection under N, which will be a maximum when $b = b_1 = \frac{a}{2}$, will in general be

$$f = \frac{Nb^2b_1^2\alpha}{3aI} \text{ inches.} \tag{17}$$

If on the other hand, $b < b_1$, the maximum deflection will occur at a distance x from the left support such that

$$x = b_1\sqrt{\frac{1}{3}+\frac{2b}{3b_1}} \text{ inches.} \tag{18}$$

Examples. 1. A cylinder liner, with $D = 25.6''$, for an explosion pressure $p_z = 356$ lbs., should, according to eq. (12n), have a wall thickness of

$$s = .047\,D = .047 \times 25.6 = 1.2 + \tfrac{3}{8} \text{ (for re-boring)} = 1\tfrac{9}{16}''.$$

With reference to Fig. 126, let $a = 50.3''$, $b = 23.6''$, $b_1 = 26.7''$. Then the maximum bending moment, due to $N = \frac{1}{10}P_z = \frac{1}{10} \times 280 \times 25.6^2 = 18400$ lbs., will be

$$M_b = \frac{18400 \times 23.6 \times 26.7}{50.3} = 230000 \text{ in.-lbs.}$$

Now the section modulus is

$$W = .8\,D_m^2 s,$$

in which D_m = mean diameter $= 25.6 + 1\frac{9}{16} = 27.2''$. Hence

$$W = .8 \times 27.2^2 \times 1.56 = 923.$$

The maximum bending stress therefore is

$$\sigma_b=\frac{230\,000}{923}=250 \text{ lbs. per sq.in.}$$

The cylinder liner proportioned according to eq. (12n) seems therefore of ample strength to take care of N_1, but the maximum deflection f should next be determined.

The moment of inertia of the cross-section is

$$I=.05(D_0{}^4-D^4)=.05[(25.6+2\times1\tfrac{9}{16})^4-25.6^4]=12\,450 \text{ in.}$$

in which D_0=outside diameter of the liner.

Taking the extension coefficient of cast iron as $\alpha=\frac{1}{11\,000\,000}$ the maximum deflection according to eq. (17) will be

$$f=\frac{18\,400\times23.6^2\times26.7^2}{3\times50.3\times12\,450\times11\,000\,000}=.00036''.$$

Since the load N is not concentrated but distributed over the length l, which makes f still smaller, this result shows that this cylinder liner is safe without any further stiffening, provided the two points of support, A_l and A_r, are not subject to deflection.

The flange or collar of the cylinder liner is very often subject to stresses, induced by the drawing up of the cylinder head studs, which tend to break the flange along a section x-y, Fig. 127. The pressure due to this drawing up must in all cases be greater than the maximum pressure on the cover, which is equal to $.785\,D_f^2 p_z$, in order to retain at the moment of explosion a sufficiently high pressure P_f on the packing surfaces to prevent them from leaking. Just how much the total pressure under the cover nuts should exceed the axial internal cover pressure for the purpose indicated, cannot be stated off hand. The kind of packing, rigidity of the cover, and the distance from packing to stud circle, all have an influence on this question, so that in favorable cases an addition (t) to the internal cover load of 5% will be sufficient, while in very unfavorable cases 50% may not be enough. With good design, practice has shown that an addition of t=from .1 to .2 to the internal cover pressure is sufficient to maintain tight joints.

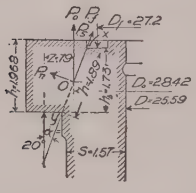

FIG. 127.—Dimensions in Inches.

The pressure on the joint due to tightening up of bolts is then

$$P_a=(1+t)p_z\times.785\,D_f^2\geqq1.1p_z\times.785\,D_f^2 \text{ pounds.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (19)$$

The flange must be strong enough to resist this static load P_a. The internal cover pressure does not enter the problem, because in cases where a separate cylinder liner is employed, the axial cover load is taken care of by the jacket wall.

The force thus developed, induces stresses in the cross-section x–y as follows:

A bending moment due to P_a and the arm z;
A tensile stress due to the component $P_n=P_a\sin\alpha$, and
A shearing stress due to the component $P_s=P_a\cos\alpha$.

The shearing force P_s should not be neglected, because the arm z of the force P_a is usually small in comparison with the height h of the cross-section x-y. The various stresses are as follows:

From the bending moment $M_b = P_a z$ and the section modulus $W = \frac{\pi D_0 h^2}{6}$ of the section, we have the maximum bending stress

$$\sigma_b = \frac{M_b}{W} = \frac{6P_a z}{\pi D_0 h^2} \text{ lbs. per sq.in.} \quad \dots \dots \dots \dots \quad (20)$$

From the normal force P_n and area of dangerous section x–y equal to $f = \pi D_0 h$, the tensile stress is

$$\sigma_z = \frac{P_n}{f} = \frac{P_a \sin \alpha}{\pi D_0 h} \text{ lbs. per sq.in.,} \quad \dots \dots \dots \dots \quad (21)$$

and from the shearing force P_s together with area f, the shearing stress is

$$\tau = \frac{P_s}{f} = \frac{P_a \cos \alpha}{\pi D_0 h} \text{ lbs. per sq.in.} \quad \dots \dots \dots \dots \quad (22)$$

Combining first, $\sigma_b + \sigma_z = \sigma_1$, and then this with τ according to Bach's equation (see p. 168, Table 15) gives finally the resultant stress,

$$\sigma_{\text{res.}} = .35\sigma_1 + .65\sqrt{\sigma_1{}^2 + 4(\alpha_0 \tau)^2}. \quad \dots \dots \dots \dots \quad (23)$$

This must not exceed the allowable bending stress K_b. For cast iron, for which the allowable tensile and shearing stresses may both be taken equal to $K_n = K_s = 2800$ lbs. per sq.in., the value of the factor α_0 is

$$\alpha_0 = \frac{K_n}{1.3K_s} = \frac{2800}{1.3 \times 2800} = .77.$$

Example. 2. The cylinder liner computed in Example 1 has the flange sketched in Fig. 127. With the dimensions there given and an explosion pressure p_z assumed at 356 lbs. per sq.in., we shall have the following:

$$P_a = 1.1 \times 356 \times .785 \times 27.2^2 = 220\,000 \text{ lbs.};$$

$$M_b = 220\,000 \times .79 = 174\,000 \text{ in.-lbs.};$$

$$P_n = 220\,000 \times \sin 20° = 220\,000 \times .342 = 75\,300 \text{ lbs.};$$

$$P_s = 220\,000 \times \cos 20° = 220\,000 \times .94 = 207\,000 \text{ lbs.};$$

$$f = 89.2 \times 1.89 = 169 \text{ sq.in.}; \qquad W = \frac{89.2 \times 1.89^2}{6} = 53.3;$$

$$\sigma_b = \frac{174\,000}{53.3} = 3260 \text{ lbs. per sq.in.}; \qquad \sigma_z = \frac{75\,300}{169} = 446 \text{ lbs. per sq.in.};$$

$$\tau = \frac{207\,000}{169} = 1225 \text{ lbs. per sq.in.}; \qquad \sigma_b + \sigma_z = \sigma_1 = 3706 \text{ lbs. per sq.in.}$$

Hence combined stress is, from eq. (23),

$$\sigma_{\text{res.}} = .35 \times 3706 + .65\sqrt{3706^2 + 4(.77 \times 1225)^2} = 4000 \text{ lbs. per sq.in.}$$

For the determination of the stress in the dangerous sections $x-y$, of the flanges of cylinders under internal pressure, Professor Bach[1] gives the following general equation:

$$\frac{3\mu R_b p z}{h_b^2} \geqq K_b. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (24)$$

Here μ is a correction coefficient which Bach determined as .8 on tests of similar cast-iron flanges, $R_b = \frac{D_0}{2}$, $p = p_z$, and z and h_b are according to Fig. 127.

If this equation had been used in Example 2, the result would have been

$$\sigma_{\text{res.}} = \frac{3 \times .8 \times 14.2 \times 356 \times .79}{(1.73)^2} = 3200 \text{ lbs. per sq.in.}$$

The first method of computation therefore gives a somewhat greater factor of safety, which, in this important place, is quite desirable.

If the flange is cored out for water passages, the amount of cutting away must be taken into account by subtracting it from the circumference πD_0 of the gravity circle. The height h under such circumstances, is usually increased so that the flange wall separating any two water spaces becomes a radial rib or web which materially assists temperature equalization. (See Figs. 119 and 120, p. 122.)

2. **The Jacket Wall** is in the direction of its axis under a tensile stress

$$P_z = p_z \times .785\, D^2 \text{ lbs.}$$

Its full cross-sectional area is expressed by (see Figs. 128 or 129)

$$f = .785(D_a^2 - D_i^2) = \pi D_m s' \text{ sq.in.}$$

The tensile stress therefore is

$$\sigma_z = \frac{p_z \times .785\, D^2}{.785(D_a^2 - D_i^2)} = \frac{p_z D^2}{D_a^2 - D_i^2}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (25)$$

or,

$$\sigma_z = \frac{p_z \times .785\, D^2}{\pi D_m s'} = \frac{p_z D^2}{4\, D_m s'}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (25a)$$

From (25*a*)

$$s' = \frac{p_z D^2}{4 D_m K_z}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (26)$$

If K_z = allowable tensile stress $\leqq$ 1800 lbs. per sq.in., and p_z is assumed = 356 lbs. per sq.in., the thickness of the jacket wall will be

$$s' \geqq \frac{356 D^2}{4 \times 1800\, D_m} = \frac{D^2}{20\, D_m}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (26n)$$

In practice s' should not be less than $\frac{7}{16}''$ for small engines nor less than $\frac{9}{16}''$ for the larger sizes. The cross-sections of the jacket wall weakened by cleaning openings or otherwise should be rechecked for strength. In the case of one-piece cylinders the tightness of the cylinder wall proper is often tested by hydraulic pressure. If the test pressure employed is equal to say $1.5 p_z = 650$ lbs., and the allowable bending stress is taken at 9250 lbs. per sq.in. for a single application of such test, the jacket wall thickness necessary to satisfy these requirements will, from eq. (12), p. 124, be

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[1] Bach, Maschinen-Elemente, 8 ed., p. 691.

$$s' \sim .25\ D_a \text{ in.} \quad . \ . \ . \quad . \ . \ . \ . \ . \ . \ . \ (26a)$$

The thrust of the connecting-rod tends to bend the jacket wall as it does the cylinder wall, and the stresses and deflections involved can be determined in the manner already shown. For cylinders which are entirely supported in the frame, the axial tensile force P_z (see pp. 103 and 113), together with the practical requirements of the foundry, will call for a thickness of jacket wall sufficient to take care of N. A closer investigation of the influence of N must, however, be made in the case of large and long stroke engines having overhung cylinders, because in this case one end of the jacket wall must take up the entire bending moment. The dangerous section is then close to the connecting flange, and in the case of weak frames it may even be found in the flange seat of the frame itself.

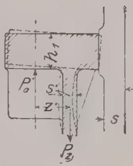


FIG. 128.

The flange used to connect the cylinder, or the jacket wall, with the frame has a metal to metal seat and is not called upon to resist gas pressure against leakage. Its dimensions therefore depend entirely upon the axial cover pressure $P_z = p_z \times .785\ D^2$, if the normal pressure N does not enter the problem. The stresses in the dangerous section of this flange may be found in the same manner as for the cylinder barrel flange, pp. 126 and 127. Bach's equation (24) there given may be written (see Fig. 128)

$$\mu P_a z' \leqq \tfrac{1}{6} \times 2\pi R_b h_1{}^2 K_b,$$

so that the maximum bending stress will be

$$\sigma_b = \frac{3\mu P_a z'}{\pi R_b h_1{}^2} = \frac{\mu P_a z'}{1.047 R_b h_1{}^2} \leqq K_b. \quad . \ . \ . \ . \ . \ . \ . \ . \ . \ . \ (27)$$

The bending moment $P_a z'$ tends to deform the flange, as shown in Fig. 128, and it may happen that the inner cylinder or liner is thereby deformed in the bore. Sufficient rigidity is therefore of great importance. This may be obtained either by increasing the thickness, s, of the cylinder wall under the flange seat, by using radial webs, or by using heavy bosses in the flange for the bolt holes. All such stiffening should, however, be left out of account in strength computations.

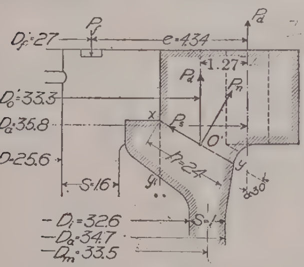


FIG. 129.—Dimensions in Inches.

The bending moment in the flange for the cylinder cover or head will be a minimum when the bolt circle coincides with the gravity circle of the cross-section of the jacket wall. In that case the axial load acts as a tensile stress only, and when two-piece cylinders are used only that part of the jacket flange lying within the bolt circle is under a bending moment due to the pressure P_f necessary to keep a tight joint. At the moment of explosion, however, this pressure P_f is relieved by the load on the cover $= p_z \times .785\ D_f^2$, so that at this instant the stress remaining is only that due to the necessary excess of P_f over the internal load on the cover.

The stresses are much more severe when the diameter of the bolt circle is greater than the outer diameter of the jacket wall. The flange is then under a heavy load due to the drawing up of the cover studs, which load induces bending, shearing, and tensile stresses, as was shown from the liner flange on p. 126. The following example, in connection with Fig. 129, explains the conditions then existing in detail.

Example. The cylinder liner, Fig. 126, computed in the previous example, is seated in the jacket flange, at the head end, as shown in Fig. 129. The pressure P_a required on the joint was on p. 127 determined at 220000 lbs. From this we can derive at once

$$\begin{aligned} P_n &= 220000 \cos 30° = 220000 \times .866 = 190500 \text{ lbs.}; \\ P_d &= P_a = 220000 \text{ lbs.}; \\ P_s &= 220000 \sin 30° = 220000 \times .5 = 110000 \text{ lbs.} \end{aligned}$$

The area of the dangerous section $x\text{–}y = f = 33.3 \times \pi \times 2.44 = 255$ sq.in., and its section modulus is

$$W = \tfrac{1}{6} \times 33.3 \times \pi \times 2.44^2 = 104.0.$$

Bending moment $M_b = P_d z = 220000 \times 1.27 = 282000$ in.-lbs.;

$$\tau = \frac{110000}{255} = 430 \text{ lbs. per sq.in.}; \qquad \sigma_2 = \frac{190500}{255} = 750 \text{ lbs. per sq.in.};$$

$$\sigma_b = \frac{282000}{104.0} = 2700 \text{ lbs. per sq.in.}; \qquad \sigma_1 = \sigma_2 + \sigma_b = 750 + 2700 = 3450 \text{ lbs. per sq.in.}$$

Hence the combined stress is

$$\sigma_{\text{res.}} = .35 \times 3450 + .65\sqrt{3450^2 + 4(.77 \times 430)^2} = 3590 \text{ lbs. per sq.in.}$$

In this case, therefore, P_s or τ could have been neglected.

The cross-section $x\text{–}y'$, should next be checked. The shearing stress for this section is found to be

$$\tau = \frac{220000}{31 \times \pi \times 1.57} = 1440 \text{ lbs. per sq.in.}$$

The combined stress in $x\text{–}y$ is close to the allowable limit.

In one-piece cylinders it is often possible to make the radius of the cover stud circle equal to the mean radius of the jacket space, or to make it so that the studs are close to the inner jacket wall. Under these conditions the bending moment of the flange is very small. An arrangement of this kind combined with sufficient rigidity for the frame flange can, however, only be had in the smaller sizes of engine.

Where special lightness of machine is desired the jacket is sometimes made out of sheet steel fitted over the head and crank-end flanges of the cylinder. This construction is not easily damaged by freezing of the cooling water, and it also admits of thorough cleaning of the jacket space from scale and mud. It has the disadvantage, however, that the steel, especially at the ends, is apt to rust through quickly, a process which is only temporarily arrested by painting or tinning.

The *connecting bolts* between frame and cylinder may be of wrought iron, if well made. The allowable stress may be taken at 5000 lbs. per sq.in. of cross-section, as measured at the bottom of the thread. It should be remembered in this connection that a few strong bolts are better than a larger number of correspondingly smaller tensile strength. The smallest number of bolts should be four for small engines. English designers, even for the larger sizes, often use only two bolts, which are placed in the horizontal bisecting plane, that is, in a position not correct with reference to connecting-rod thrust. In general, the less the apparent rigidity offered by the design of the cylinder and frame, the greater should be the number of connecting bolts used.

For cylinder-head packings and cover studs, see pp. 137 and 138.

III. Cylinder Covers (Cylinder Heads) and Stuffing Boxes

Material is usually tough, close-grained cast iron. During the first years of the development frequent cylinder-head fractures drove the designer to the use of cast steel. Since then, however, much simpler forms for the heads, which can with perfect safety be made of cast iron, have been developed.

The *allowable stress* should, in such places where it can be determined with any accuracy at all, be kept moderately high, for cast iron say $K_b = 3500$ lbs. per sq.in. This is done in order to get a construction not too rigid against expansion by heat and to obtain better conduction of heat through the walls. In general, however, the less the certainty with which the acting forces can be determined, the lower should be the value of K_b.

The distinction usually made between the term *cylinder cover* and *cylinder head* is that the former has a shape approximating a flat plate, or a plate having but simple curvature. The latter, on the other hand, contains the valves, and its shape is therefore much more complex. In fact in some cases the dimensions of these heads in the axial direction is so great that they practically contain the combustion chamber. The triple duty (cover, valve cage, and combustion chamber) so imposed makes the cylinder head a machine member of the greatest importance and subject to the most severe stresses. Its design, for machines above a limited size, is further complicated by the fact that it is difficult to apply an even fairly rigid method of computation, and everything depends upon feel and judgment of the constructor.

Simple cylinder covers are obtained when the valves are placed at the side of the cylinder, leading into the combustion chamber. This method is, however, only possible in the smaller sizes and where moderate compression is employed. In large machines with high compression there is not enough room for the valve ports in the circumferential surface of the compression space. Besides this, these ports weaken the cylinder, prevent the use of separate cylinder liners or bushings, and form pockets in which exhaust gases are apt to remain and in which ignition is imperfect. Not quite so bad in this sense is the new arrangement of placing the valves at top and bottom at the end of the cylinders. But even this scheme is open to some objections, as, for instance, the breaking up of the combustion chamber. It is also to be feared that, although by this design the cylinder heads are relieved and made much simpler in construction and safer in operation, the trouble has only been transferred to the cylinder, itself a machine member which already requires the most careful consideration in its design.

The design of the cylinder head is in the main subject to the following three **fundamental requirements.**

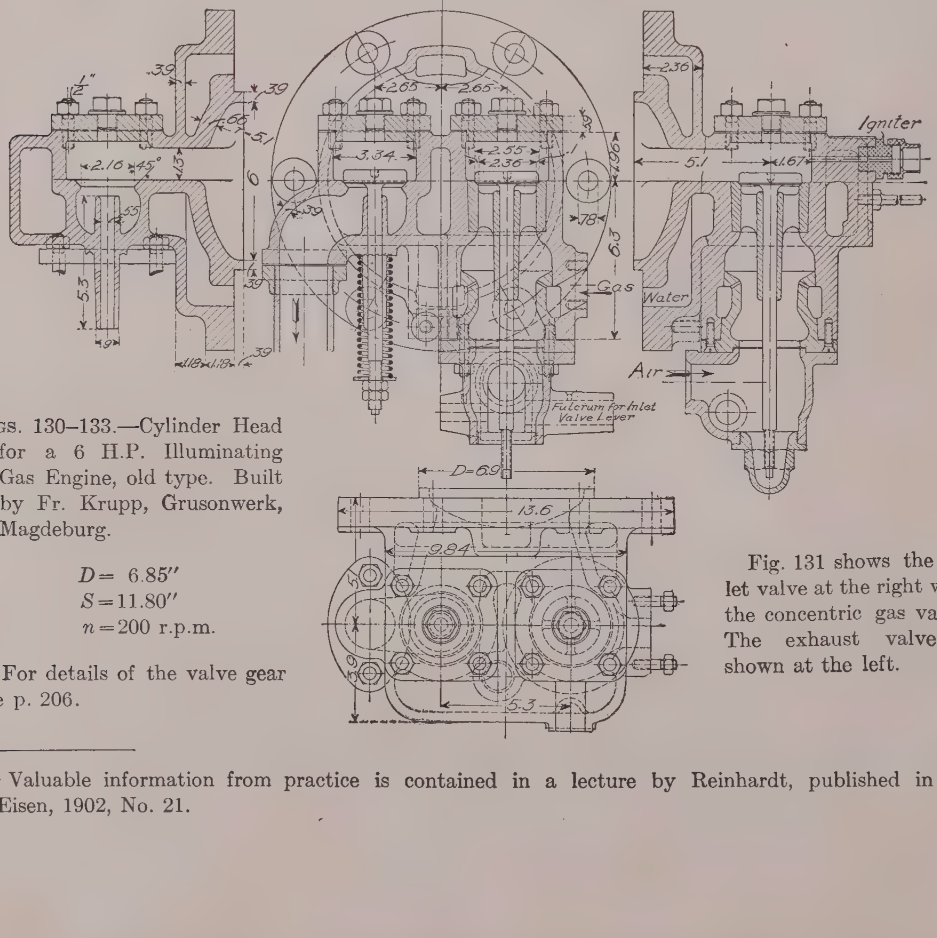
(*a*) *Thermodynamically proper form of the combustion chamber*, so that the distances for flame propagation in all directions from the igniter shall be the smallest possible, that there shall be no dead spaces or corners and inadequately cooled projections, and so that the movement of the gases from the combustion chamber into the cylinder proper shall occur in an axial direction as closely as possible. The inner form of the compression chamber should therefore approximate a hemisphere, or a short cylinder (length = diameter). The inner walls should be unbroken and smooth, and care should be taken not to deflect the stream of gas laterally as it enters the cylinder bore (see Part V).

(*b*) *Correct arrangement of the openings* for valves and igniters. The unavoidable valve ports should reduce the enveloping surface of the chamber as little as possible. With due reference to practical requirements, these ports should be so distributed that there shall be an approximately symmetrical distribution of metal in the central cross-sections, so that the water-cooling may be equally effective on all sides. Local accumulations of heat causing unequal expansions are thereby prevented and there will be no pockets out of the path of the jacket water. The jacket wall, in proper places, should be provided with sufficiently large cleaning openings, through which the space may be cleaned very thoroughly of core sand, and any deposit by the cooling water in the way of scale or slush can be removed from time to time. (Failure to remove such accumulations from the cooling surface has already resulted in serious accidents in operation.)

(c) *A general shape of the cylinder head such that a little flexibility prevents the occurrence of high stresses.* The design should be as far as possible of regular form, and of sufficient strength and stiffness, but the inner and outer walls should not be so rigidly connected that the latter cannot accommodate itself to the greater expansion by heat of the former. (Compare p. 120.) Unnecessary piling up of metal and abrupt changes in the thickness of cross-section should therefore be carefully avoided, the two walls should be connected only where necessity demands it, and rigidity should be provided for only in places where it is required (flanges or valve seats).

The third requirement above outlined is, of all things connected with gas-engine design, the most difficult to properly carry out. The designer is thrown entirely upon judgment and trial, and even the most experienced are not safe against failure. The construction of a large cylinder head, more than that of any other machine member, therefore calls for close co-operation of drafting room and foundry, but even then the most careful work in the casting of the first few heads of new design does not always guarantee immunity against fracture in operation.[1]

Designs of Cylinder Covers and Heads. (See also Figs. 71, 124.)



Figs. 130–133.—Cylinder Head for a 6 H.P. Illuminating Gas Engine, old type. Built by Fr. Krupp, Grusonwerk, Magdeburg.

$D = 6.85''$
$S = 11.80''$
$n = 200$ r.p.m.

For details of the valve gear see p. 206.

Fig. 131 shows the inlet valve at the right with the concentric gas valve. The exhaust valve is shown at the left.

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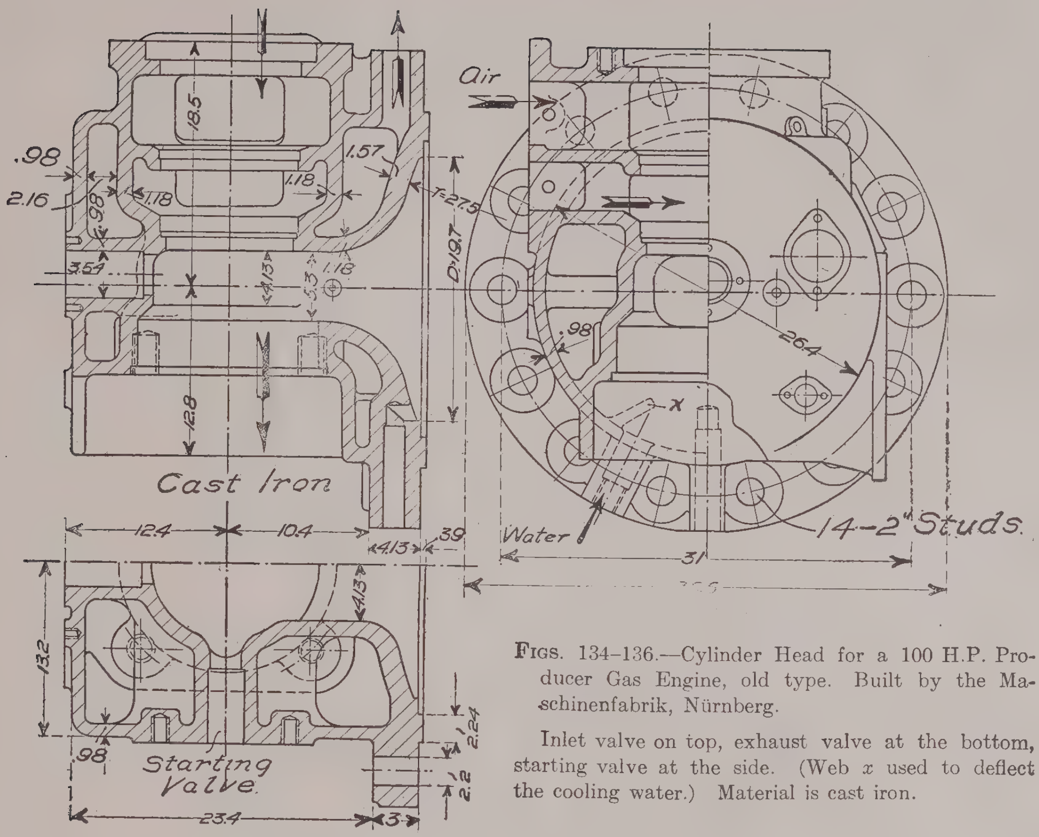
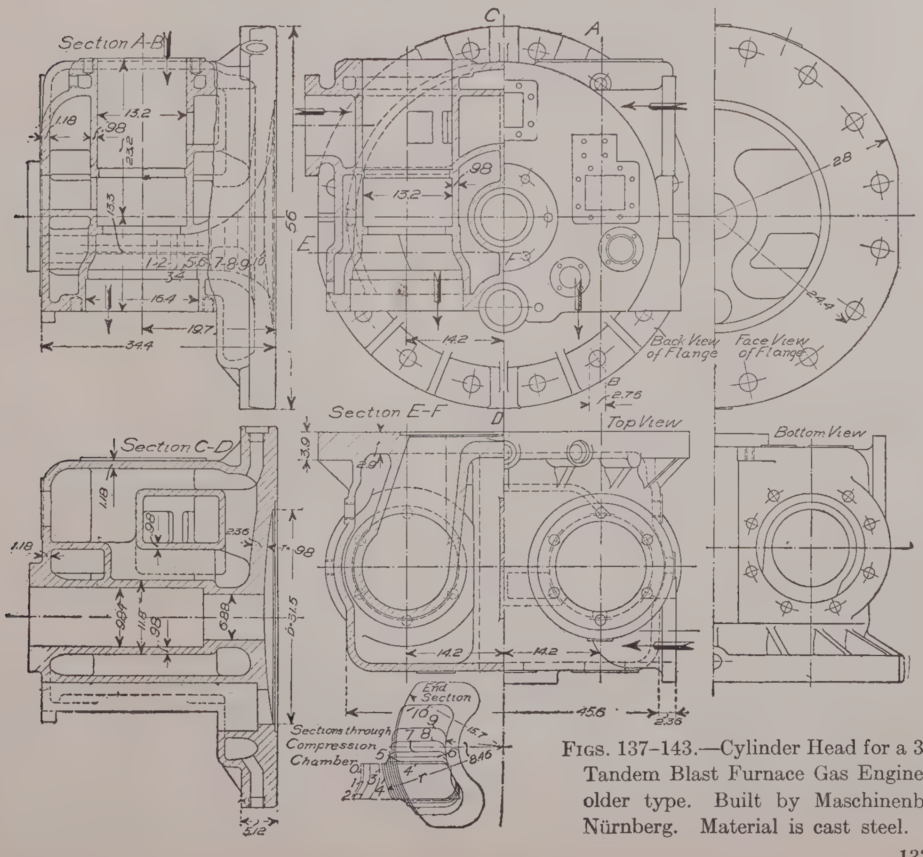
[1] Valuable information from practice is contained in a lecture by Reinhardt, published in Stahl and Eisen, 1902, No. 21.

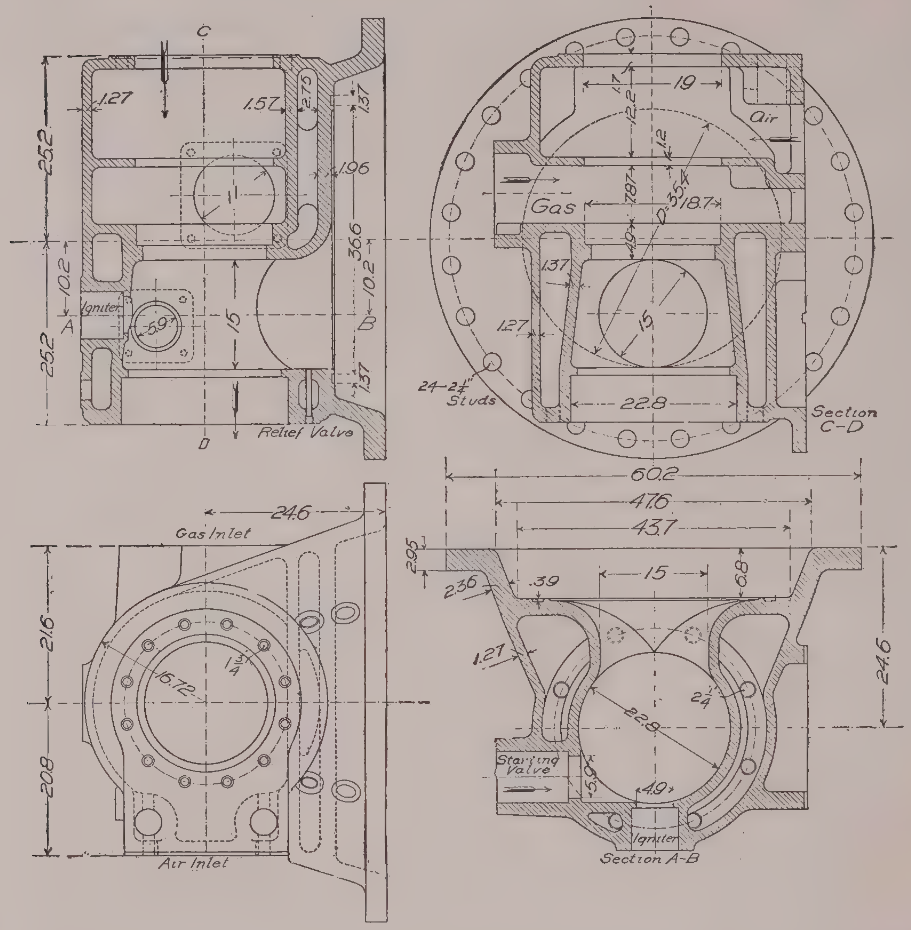


Figs. 134–136.—Cylinder Head for a 100 H.P. Producer Gas Engine, old type. Built by the Maschinenfabrik, Nürnberg.

Inlet valve on top, exhaust valve at the bottom, starting valve at the side. (Web x used to deflect the cooling water.) Material is cast iron.



Figs. 137–143.—Cylinder Head for a 300 H.P. Tandem Blast Furnace Gas Engine, of the older type. Built by Maschinenbau-Ges, Nürnberg. Material is cast steel.



FIGS. 144–147.—Cylinder Head for a 300 H.P. Gas Engine built by L. Soest & Co., Reisholz, Düsseldorf. Material is cast iron. (The Frame for this engine is shown in Figs. 87–89, p. 107).

Constructive Details. Strength computations give useful results only in the case of the simpler covers. As regards cylinder heads no close mathematical treatment is possible on account of complexity of form and uncertainty of casting stresses. The designer is forced to rely upon experienced judgment and a few tried practical rules.

The cylinder cover of simplest form is a plain circular plate, which may be considered as a beam under transverse stress, supported at the ends of a diameter. From what has preceded (p. 126), the cover-nuts must be drawn up tightly enough, so that at the moment of greatest pressure p_z enough pressure still remains on the joint to keep it tight. If, as before, we assume the total stress in all of the studs to be

$$P_d = (1+t)p_z \times .785\,D_f^2 \text{ lbs.},$$

then each one of i studs has to carry a load of

$$P = \frac{P_d}{i} = \frac{(1+t)p_z \times .785\,D_f^2}{i} \text{ lbs.} \quad \ldots\ldots\ldots\ldots \quad (28)$$

The pressure P_d produces a constant bending moment

$$M_b=\frac{P_d e}{\pi}=\frac{iPe}{\pi}\text{ in.-lbs.}, \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad (29)$$

which leads to the requirement that

$$WK_b \geqq \frac{\mu P_d e}{\pi}=\frac{\mu iPe}{\pi}. \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad (30)$$

In these equations e is the radial distance from center line of the stud to the middle of the packed surface (see Fig. 129, p. 129), and μ a correction coefficient which, according to Bach,[1] may be made $=\frac{6}{5}$ for cast iron. If the cross-section of the cover is of simple rectangular shape, without water jacket, whose dimensions are, thickness $=h$, and diameter (length) $=D_d=2R_d$, the section modulus of the central section will be

$$W=\tfrac{1}{6}D_d h^2=\tfrac{1}{3}R_d h^2.$$

We may then write

$$K_b\times\tfrac{1}{3}Rh_d^2\geqq\frac{\mu P_d e}{\pi}=\frac{\mu iPe}{\pi},$$

and consequently the bending stress in the section, independent of the explosion pressure, p_z and depending only upon the stress P_d in the studs is, putting $\mu=\frac{6}{5}$,

$$\sigma_b=\frac{3.6P_d e}{R_d h^2\pi}=\frac{3.6\,iPe}{R_d h^2\pi}\text{ lbs. per sq.in.} \quad . \; . \; . \; . \; . \; . \; . \; . \quad (31)$$

At the moment of explosion the internal load on the cover is $p_z\times.785D_f^2$ lbs., which tends to bend the cover outward, and thus decreases the pressure P_f upon the packing to a remainder $tP_f=iP_0$, where $t<1$ represents the percentage of excess pressure as before explained (p. 129). The maximum total load on the cover is therefore

$$P_f=(1+t)p_z\times.785\,D_f^2=iP_0+p_z\times.785\,D_f^2\text{ lbs.} \quad . \; . \; . \; . \; . \; . \; . \quad (32)$$

Of this amount the force $\frac{tP_f}{2},=\frac{iP_0}{2}$, acts along the semi-circumference $\pi\frac{D_f}{2}=\pi R_f$; while the load $\frac{.785\,D_f^2p_z}{2}$ is uniformly distributed over the semi-circular area $\frac{\pi R_f^2}{2}$. For the purpose of simplification, however, we will assume that both forces act at the center of gravity of the semi-circle, that is, at an arm $a=\frac{4}{3}\frac{R_f}{\pi}=.4244\,R_f$ from the center of the cover. The maximum bending moment then is

$$M_b'=\frac{(1+t)\,p_z\times.785\,D_f^2}{2}\times.4244\,R_f\text{ in.-lbs.}, \quad . \; . \; . \; . \; . \; . \; . \quad (33)$$

or, since $D_f=2R_f$, so that $.5\,D_f^2=2\,R_f^2$, and $.4244\times.785=\frac{1}{3}$, we may write,

$$M_b'=\frac{(1+t)p_z\times2\,R_f^2R_f}{3}=\frac{2}{3}(1+t)\,p_zR_f^3\text{ in.-lbs.} \quad . \; . \; . \; . \; . \; . \; . \quad (33a)$$

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[1] Bach, Maschinen-Elemente, 8 ed., p. 37.

But it has been shown above that the section modulus of a plane cover is

$$W=\tfrac{1}{3}R_d h^2.$$

Hence the bending stress resulting from M_b' will finally be

$$\sigma_b=\frac{2(1+t)p_z R_f^3}{R_d h^2}\text{ lbs. per sq.in.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (34)$$

Bach,[1] in his computations in this case, proceeds as follows:
Combining the moment of the internal pressure

$$=.5\,\pi R_f^2 p_z\times\frac{4}{3}\frac{R_f}{\pi}=\frac{2}{3}R_f^3 p_z,$$

the moment of the pressure remaining on the packing at the instant of explosion

$$=.5 iP_0\frac{2R_f}{\pi}=\frac{iP_0 R_f}{\pi},$$

and the moment of the total stress $iP_0+\pi R_f^2 p_z$ in the studs

$$=.5\,(iP_0+\pi R_f^2 p_z)\frac{2\,R_d}{\pi},$$

the resulting bending moment will be

$$M_b'=(iP_0+\pi R_f^2 p_z)\frac{R_d}{\pi}-\frac{2}{3}R_f^3 p_z-iP_0\frac{R_f}{\pi}$$

$$=R_f^2 R_d p_z-\frac{2}{3}R_f^3 p_z+iP_0\frac{e}{\pi}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (35)$$

If the distance e, see Fig. 129, from stud circle to middle of packing is small, the pressure $iP_f=iP_0$ will also be small. This is to be aimed at if possible. If this pressure is neglected, and $\mu=\frac{5}{6}$ is introduced into eq. (35), this may be written

$$M_b'=\tfrac{6}{5}\times R_f^2 R_d p_z-\tfrac{2}{3}R_f^3 p_z\leqq WK_b. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (36)$$

The maximum bending stress for the rectangular cross-sections is then

$$\sigma_b=\frac{\frac{6}{5}R_f^2 R_d p_z-\frac{2}{3}R_f^3 p_z}{\frac{1}{3}R_d h^2}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (37)$$

In order to simplify eq. (37), Bach substituted for R_f (the radius up to the middle of the packing), the stud circle radius $R_d=R_f+e$. Then

$$M_b=\tfrac{1}{3}R_d^3 p_z\text{ in.-lbs.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (38)$$

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[1] Bach, Maschinen-Elemente, 8 ed., p. 683.

A simple flat cast-iron cover therefore requires a thickness

$$h \geqq R_d \sqrt{\frac{6}{5} \frac{p_z}{K_b}}, \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad (39)$$

or, putting $p_z = 350$ lbs., and $K_b = 3500$ lbs. per sq.in.,

$$h = R_d \sqrt{\frac{6 \times 350}{5 \times 3500}} = .346 R_d \sim \tfrac{1}{3} R_d \text{ ins.} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad (39n)$$

Since, however, especially in case separate cylinder liners are employed, the distance e is apt to be considerable, it is recommended that eq. (34) or (37) be used.

For covers which are water-jacketed, the moment of inertia of the dangerous section, as far as this can still be determined, must be found, and from this the section modulus $W = \frac{I}{e}$ is found in the ordinary way. If the cover is pierced by only a single and comparatively narrow axial opening, it may be assumed in the computations that the connection between outer and inner walls is sufficient to overcome any weakening due to the opening, i.e., both walls may be considered unbroken, since experience has shown that a fracture never passes through the hole, but is always found outside of the connecting wall.[1]

But if there are in the water-cooled cover several irregularly placed chambers, or if there is a comparatively large central opening, computation fails and estimating has to be resorted to. For cylinder heads in their complicated forms this latter method of design is at any rate the most common. Generally reliable practical rules cannot be obtained in this case even from designs successful in practice, as too many factors enter the problem. If the cylinder head has a free-connecting flange, which is the case in at least all of the larger machines, special attention should be paid to the transition from this flange to the cylindrical surfaces of the head. The distance e above explained, see Fig. 129, may be so great that considerable bending stresses are set up at this point. It is usually possible to satisfactorily check this flange by itself, by the method outlined on p. 126. A second cross-section under severe stress is usually formed through the center planes of the valve seats, especially if the valves are large, and arranged concentrically one above the other. The danger of fracture at this point is emphasized materially when the valve cages have conical form, the seats strongly pressed together having a tendency to cause fracture by wedge action.

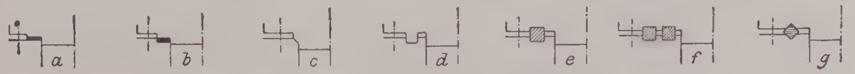

FIG. 148.—Methods of Packing Cylinder Heads.

The inner surfaces of the heads should be curved if possible. This not only adds to the strength but is also favorable to free expansion and to combustion. Large internal flat surfaces are dangerous. In determining the thickness of the walls any external load brought to bear by valve housings, valve gear, pipe connections, etc., should not be forgotten.

Fig. 148 *a–g* shows several forms of **cover packings.** Of these the first three can be used only in the smaller sizes of engines. In general, only those forms of packing in which the packing material cannot be driven outward by the gas pressure offer sufficient safety. The simple tongue and groove arrangement, Fig. 148 *d*, is probably the most common, but costs more and is not so easy to make as the loose-packing ring, Fig. 148 *e–g*. Such a ring can first be accurately made in itself and then easily ground to a fit on the packing surfaces of cylinder and head. Between the surfaces of the ring and the flange it is usual to place a thin sheet of asbestos paper. A thick layer of such material, much above $\frac{1}{32}''$, should not be used because the studs have to be drawn up too tightly to keep it from blowing out or to keep the gases from breaking through it. It is evident that to obtain a good joint it is necessary in the first place to machine these surfaces as accurately as possible, that is, to produce a metallic joint as far as practicable, and then to use only material of good heat-resisting quality, as asbestos, vulcanized fiber, etc., for the packing.

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[1] Compare also Bach, Maschinen-Elemente, 8 ed., p. 685.

After many years of failure, forms of **stuffing-boxes** have lately been developed which completely satisfy the requirements of present day operation. It has been shown that only cast iron is suitable for the packing rings, and that all of the softer alloys will not stand the wear. The rings are given but very little spring, and tightness of joint is secured by a close-fitting of the bore of the ring to the rod, and the use of a number of rings in series. A good method of forced lubrication, distributing the oil at several places along the length of the packing, is essential to good operation.

It is of advantage not to house the packing in the cylinder cover itself but to use a separate cage, and to provide this with separate water-cooling arrangements. This permits a definite control of the mean temperature of the gland. The rule should be departed from only when the head or cover is free from valve and igniter openings, thus forming a simple body with wide and unobstructed water spaces. But even then separate water cooling and forced lubrication should still be adhered to.

If a special design is desired it is well to use as a model tried forms of stuffing-boxes such as are built ready for installation by a few firms. For large gas engines the Schwabe stuffing-box, Fig. 149, is most commonly used. The dimensions of this gland necessary to fix the dimensions of the cylinder cover are given in Table 10.

Design of Stuffing-boxes.

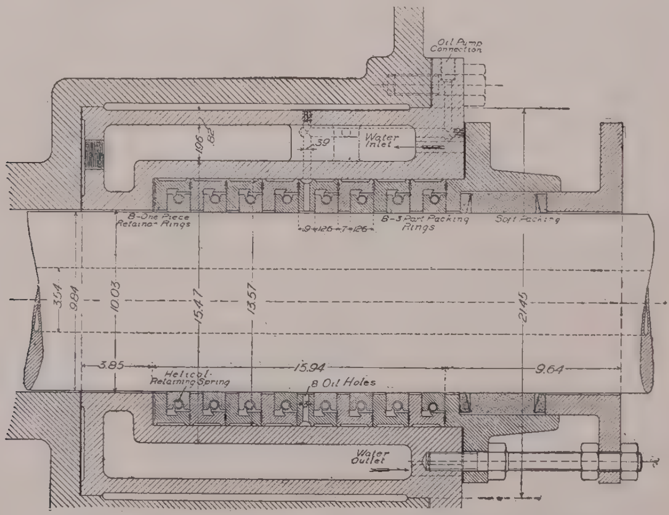

Fig. 149.—Schwabe Stuffing Box for Horizontal Engine.

TABLE 10

PRINCIPAL DIMENSIONS, IN INCHES, OF SCHWABE'S GAS-ENGINE STUFFING-BOXES

d	*a*	*b*	*c*	*e*	*f*
3.14	5.50	9.2	1.18	.79	.63
3.94	6.30	9.2	1.18	.79	.63
4.72	7.48	10.9	1.42	.95	.71
5.51	8.26	10.9	1.42	.95	.71
6.30	9.45	14.4	1.65	1.10	.79
7.09	10.22	14.4	1.65	1.10	.79
7.88	11.40	16.5	1.89	1.26	.86
8.66	12.20	16.5	1.89	1.26	.86
9.45	13.40	18.3	2.13	1.46	.86
10.22	14.15	18.3	2.13	1.46	.86
11.00	15.55	20.3	2.38	1.65	.95
11.80	16.15	20.3	2.38	1.65	.95

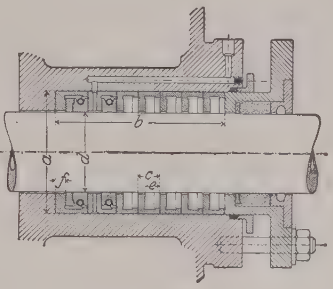

FIG. 150.

Up to $d=6''$, the normal number of rings is 7, above $6''$ to $d=12''$, 8 rings.

The **cylinder head-studs** are made mostly of best wrought iron. In many cases lack of room for the nuts compels the use of a smaller diameter of stud, when soft steel may be employed, in which case the allowable stress, based on the cross-sectional area at bottom of thread, may be taken at from 5500–6500 lbs. per sq.in.[1]

Number of studs, if cylinder head is sufficiently rigid, about

$$i = .25\,D + 4. \quad \text{(40)}$$

Many works, however, use a smaller pitch, up to

$$i = .5\,D + 4. \quad \text{(40a)}$$

D = diameter of cylinder in inches.

To protect the packing, the pitch between studs should not exceed say $7''$. Of course in case of necessity greater distances, up to $10''$, may be employed, but with such spacing, even rigid flanges do not always prevent the opening of the joint.

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[1] TRANSLATOR'S NOTE. The use of soft steel for such purposes is probably universal in this country, wrought iron being little used in machine construction.

IV. Pistons, Piston Rings and Piston Rods

Material for the piston heads, usually a good grade of close-grained cast iron; for, in the case of large engines, now and then, also cast steel. White metal lining on the bearing surfaces for both of these materials is sometimes used. For piston rings a medium hard, but not too brittle, cast iron is best. Soft steel, with or without white metal covering, is sometimes employed.

The stresses occurring can be definitely determined only in rare cases, as for instance for the face of the piston. For cast iron the allowable stress may be taken at K_b=from 5000 to 6000 lbs. per sq.in.

1. The pistons of *single-acting internal-combustion engines* are long, open trunk pistons, which, besides taking care of the gas pressure, are also called upon to carry the thrust due to the connecting-rod. In some of the later designs of large single-acting engines, the rod thrust is again transferred to a separate cross-head, and the bearing surfaces of the piston are often further relieved by carrying the weight of the piston on the cross-head and on a tail-road bearing. Quite as sound and worthy of imitation is the tendency to provide all of the larger trunk pistons with water-cooling arrangements. Water cooling was for a long time looked upon as a serious complication, not only in operation but also in construction, and for that reason was not employed except in cases of necessity. But since the advantages of cooled pistons are more clearly understood, they are now gradually coming into use for the smaller sizes of machine, down to 20″ diameter. Above 150 H.P. all engines should be provided with water-cooled pistons. The main advantages of this type of piston are to be sought mainly in greater reliability of operation and in greater durability. Cooled pistons allow of greater compression, and by changing the outlet temperature of the cooling water, give a means for controlling the play between piston and cylinder wall. They decrease friction and consequently, also, the consumption of lubricating oil, and, lastly, do away with *radiation* and prevent the formation of annoying oil vapors.

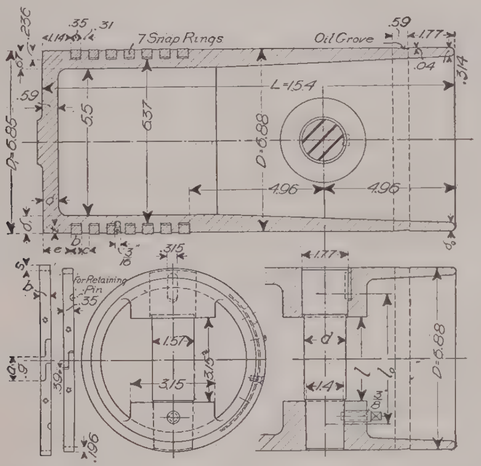
Double-acting pistons are, in general, much shorter than the trunk pistons of single-acting machines. Their length is mainly determined by the number of necessary piston rings, unless, as in the Körting 2-cycle, the piston controls ports. A length of piston greater than that required for the rings does not offer any further advantages. Excessive length increases the weight of piston and its water contents, i.e., increases the weight of the reciprocating parts, increases the load on cylinder walls, piston rods, and stuffing-boxes; complicates the manufacture and increases the difficulty of properly lubricating the piston surface. In the building of large engines the method of supporting the weight of the piston on the sides of the cylinder is being gradually abandoned. Instead, the piston is fitted in the bore with comparatively large clearance and its weight is taken up by special bearings outside of the cylinder. The office of the piston rings is then merely to make the piston gas-tight.

A well-designed system of stiffening webs is of special service in the case of double-acting pistons. This should act as an effective stiffening for the entire body of the piston, especially between the face and the rod boss, and should at the same time promote the circulation of the cooling water. The core and cleaning openings should be placed in the sides of the piston and not in the face, to prevent any burning in of fastening screws, and any weakening of the face. Cleaning openings are in any case a bad feature of piston design, and their use may be avoided by using a two-part piston, see Fig. 169, p. 142. Unless the cooling water is especially impure, a troublesome deposit of scale is of uncommon occurrence, because the rapid movement of the water, in spite of high temperature, does not allow the deposition of much scale-forming material.

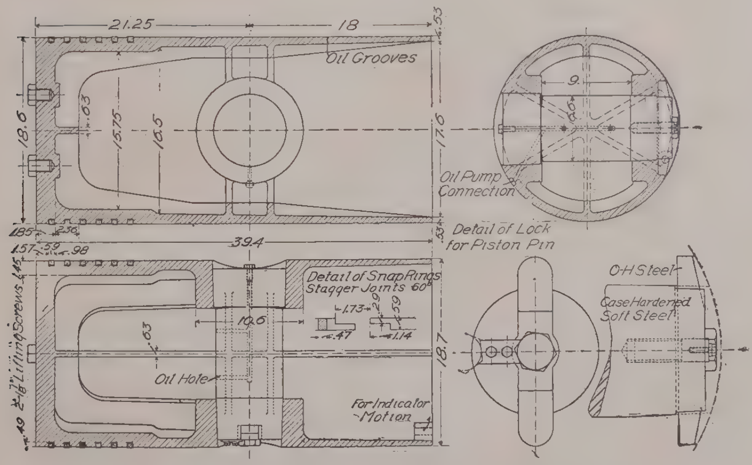
The proper fastening of the piston body on the rod offers some difficulty on account of water inlet and exit openings encountered in the rod at this place. Further, the high explosion pressure necessitates rather large connecting surfaces to resist the axial pressure, if the unit pressure on these surfaces is to be kept within reasonable limits. Various methods of solving this problem are shown in Figs. 167–170.

The fits at the ends of the piston rod for the cross-head and tail-road bearings should be from $\frac{1}{4}''$ to $\frac{3}{8}''$ smaller in diameter than the rest of the rod, in order to be able to slip on the stuffing-boxes and cylinder heads after the rod has had to be turned down on account of wear.

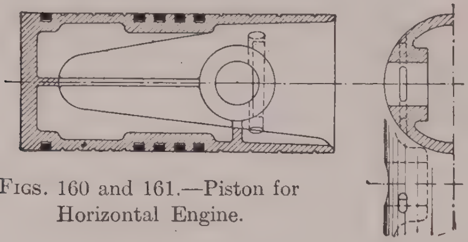
Design of Pistons and Piston Rods.



FIGS. 151–154.—Piston for a 6 H.P. Horizontal Engine.



FIGS. 155–159.—Piston for a 100 H.P. Güldner Engine.



Figs. 160 and 161.—Piston for Horizontal Engine.

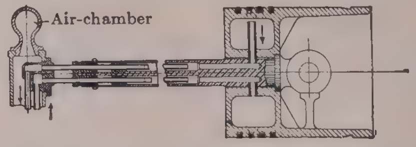


Fig. 162.—Water-cooled Trunk-piston, water inlet and outlet through the tail-rod.

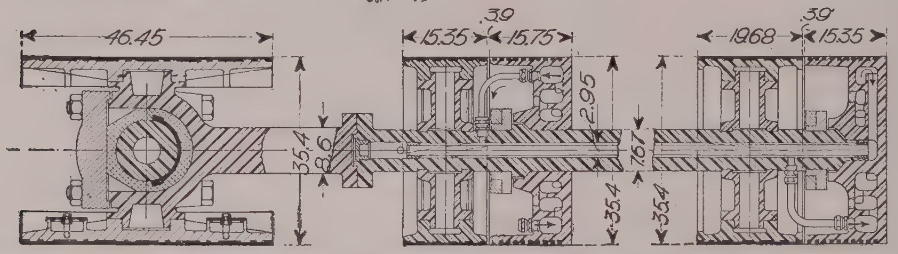
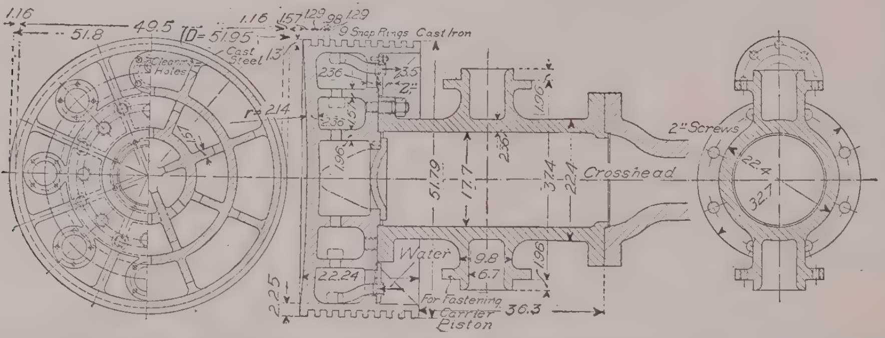
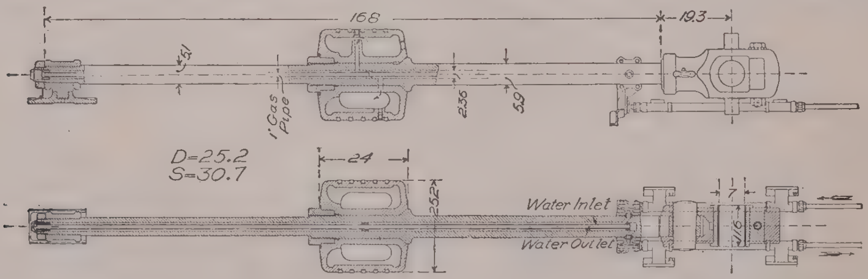


Fig. 163.—Tandem Pistons with Carrier Pistons and Cross-head, 700 H.P. Nürnberg Engine (old type).



Figs. 164–166.—Water-cooled Disk-piston, 700 H.P. Nürnberg Engine, old type. See also Fig. 163.



Figs. 167 and 168.—Piston and Piston Rod, 300 H.P. Double-acting, 4-cycle Engine, built by Gasmotorenfabrik Deutz. (See also Plates IX and X).

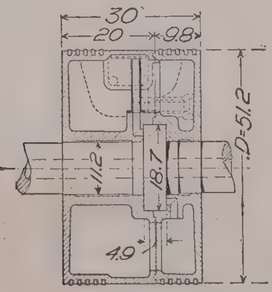


Fig. 169.—Disk Piston for 1500 H.P. Double-acting, 4-cycle Simplex Engine. (Stroke 55.2″, Assembly Drawing in Part III.)

This method of piston construction has the following advantages: the two parts can be thoroughly cleaned of core sand during construction and of scale during operation, there is no thread on the rod and no nut in the combustion space, and lastly the holding surface on the rod is normal to the pressure on the piston.

Constructive Details. The general formulas for the design of cross-heads may be applied to the design of *trunk pistons.* The length L of such pistons mainly depends upon the connecting-rod thrust, since this determines the unit bearing pressure K. We must have

$$\frac{N_{\max}}{DL} \leq K \text{ lbs.}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (41)$$

in which $N_{\max}$ is the maximum value of the connecting-rod thrust.

It can be shown, p. 158, that, for a ratio of connecting-rod to crank $=5$,

$$N_{\max} \sim \frac{P_z}{10} \text{ lbs.}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (42)$$

in which $P_z =$ maximum explosion pressure.

Hence the piston length L must be

$$L \geq \frac{.1 P_z}{KD} = \frac{p_z \times .785\, D^2}{10\, KD} = \frac{p_z \times .0785\, D}{K} \text{ ins.} \quad . \quad . \quad . \quad (43)$$

In this L may be taken to represent the real carrying length of piston, i.e., the total length of piston after subtracting the part containing the ring grooves and that part of the inner end usually turned down somewhat to allow for expansion. These amount to about one third of the total piston length. But this cutting down of bearing surface may also be taken into account by keeping the unit pressure K down to a low figure, in which case L from the above formula may stand for the entire length. This last method is the simpler of the two and will be used here.

According to the experience gained in steam-engine practice with independent cross-heads, the maximum unit pressure K should not exceed from 35 to 45 lbs. per sq.in. In view of the fact that the main office of the trunk piston is to maintain a gas-tight packing, and that on this account the bearing surface must be constantly kept in the best possible shape, the pressure K for such pistons must be kept considerably below the above figures, in spite of the fact that $N_{\max}$ occurs only every fourth stroke and that in general the total thrust is not serious. In large trunk pistons it is well not to exceed 21 lbs. per sq.in. For this pressure the *minimum length* of piston then is

$$L \geq \frac{p_z \times .0785\, D}{21} = .00374 p_z D \text{ ins.}, \quad . \quad . \quad . \quad . \quad . \quad (43)$$

and for the normal diagram, with $p_z = 356$ lbs.,

$$L \geq .00374 \times 356 D = \sim 1.3\, D.. \quad . \quad . \quad . \quad . \quad . \quad (43n)$$

In small engines in which longer pistons offer no special difficulties, L may be made as great as $2.5\, D$, that is, $K =$

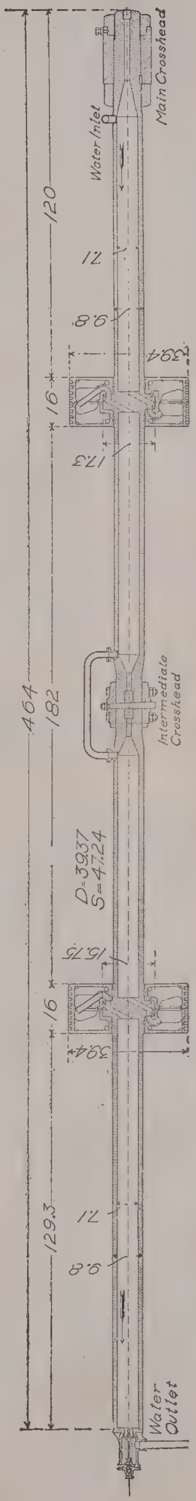


Fig. 170.—Piston and Rod for a 1200 H.P. Double-acting 4-cycle Simplex Tandem Engine. (Assembly Drawing in Part III.)

from 10–11.5 lbs. per sq.in. In large engines, however, excessive length of piston leads to the difficulties already pointed out, i.e., excessive weight of reciprocating parts, inequality in expansion by heat, trouble in proper fitting and in uniform lubrication.

The weight of uncooled pistons in horizontal machines usually does not reach .75 lbs. per sq.in. of projected piston area LD, and therefore has no appreciable influence on the value of K. Where water cooling is used, however, it may under certain conditions, so greatly increase the weight of the piston that it should be taken into account.

The thickness of the *face of the piston*, without webbing, may be found by considering it a flat round plate supported at the edge and uniformly loaded with the force of explosion p_z. The dangerous section of such a plate is found along a diameter, as is the case for simple cylinder covers. This section is under the action of two moments acting in opposite directions. The first is due to the fluid pressure $\frac{D^2\pi p_z}{8}=\frac{R^2\pi p_z}{2}$ acting at an arm equal to $\frac{4R}{3\pi}$, so that the moment is

$$M_{b_1}=\tfrac{2}{3}\,R^3p_z.$$

The second results from the reaction at the supporting edge, and is equal to the force $\frac{R^2\pi p_z}{2}$ acting at an arm of $\frac{2R}{\pi}$, so that its moment is

$$M_{b_2}=R^3p_z.$$

The resulting moment is therefore

$$M_b=R^3p_z-\tfrac{2}{3}\,R^3p_z=\tfrac{1}{3}\,R^3p_z. \qquad (44)$$

The section modulus of a flat plate of thickness, δ, is

$$W=\tfrac{1}{6}\,D\delta^2=\tfrac{1}{3}\,R\delta^2.$$

From the relation

$$M_b \leq WK_b \quad \text{or} \quad \tfrac{1}{3}\,R^3p_z \leq \tfrac{1}{3}\,R\delta^2K_b, \qquad (44a)$$

we can therefore derive the required thickness of piston face

$$\delta=R\sqrt{\frac{p_z}{K_b}}. \qquad (45)$$

Inversely, with a given value of δ, the unit stress in the section will be

$$\sigma_b=\frac{p_zR^2}{\delta^2}\leq K_b. \qquad (46)$$

[1] Batch modifies eq. (45), writing it $\delta \geq r\sqrt{\mu\frac{p}{K_b}}$, introducing a coefficient, μ, whose value for a plate simply supported at the edge is $\frac{6}{5}$, and for one rigidly fastened at the edge is $\frac{4}{5}$. In eq. (45) we have assumed the average value $\mu=1.0$, and have taken the radius of free length $R=\frac{D}{2}$ (instead of r = radius of the *inner* surface of the piston

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[1] Maschinen-elemente, 8 ed., p. 37.

face, as Bach has done). This assumption, for most conditions excessively safe, allows the use of a comparatively high value for the safe stress $K_b=7500$ lbs. per sq.in., and putting $p_z=356$ lbs., we obtain the special equation

$$\delta=R\sqrt{\frac{356}{7500}}=.22\,R. \qquad \text{(45}n\text{)}$$

In case the piston is webbed, the proper section modulus, instead of $\frac{1}{6}D\delta^2$, is to be used in eq. (44*a*). But care should be taken not to overrate the strengthening effect of such webs, on account of additional temperature and internal stresses and the effect of the thrust N, none of which have been allowed for in the above discussion. That such overrating is possible is sufficiently shown in Fig. 171, which represents the face of the piston of a large marine engine (Loutzki), which was completely shattered by an explosion much more severe than the ordinary. This occurred when, during the trial of an engine operating on a new constant-pressure cycle with a maximum pressure of about 350 lbs., the supply of oil, owing to some irregularity, became such as to suddenly change the slow combustion into an explosion, during which a maximum pressure of about 1400 lbs. per sq.in. seems to have been generated. An occurrence of this kind, however, is much less dangerous to the attendants, and less costly to repair than the fracture of a cylinder head for instance, and for that reason it is not a bad idea to purposely design the piston face weaker than the cylinder cover and thus to constitute it a sort of safety-valve.

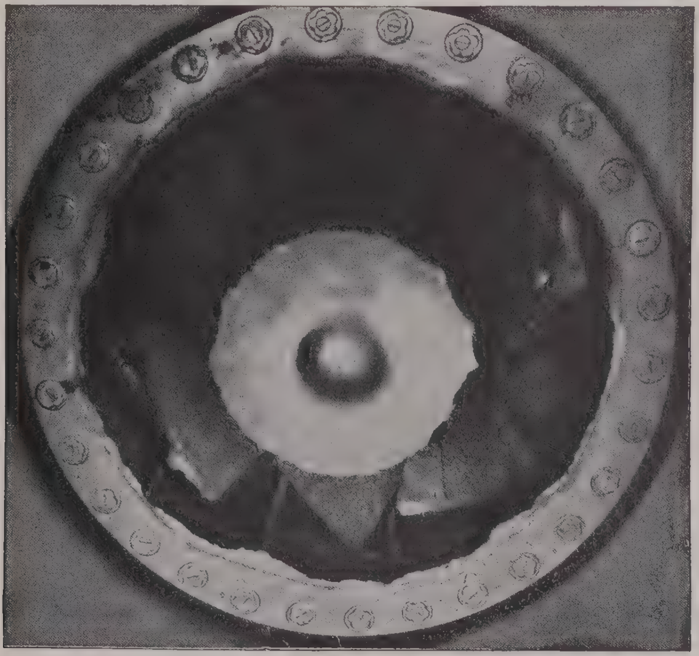

Fig. 171.—Fractured Piston Head, Experimental Engine, 27.5″ Cyl. Dia.

The maximum thickness of the *piston barrel* (for meaning of symbols see Figs. 151–154, p. 141) may be made about

$$\delta_1=\left(\frac{D}{50}+t+.2\right) \text{ ins.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (47)$$

The decrease of δ_1 toward the open end depends upon the system of webbing used, the position of the rings and of the wrist pin, and upon considerations of manufacture. The thickness at the open end may, if the webbing is sufficient, be made

$$\delta_0=\tfrac{1}{2}\delta_1 \quad \text{to} \quad \tfrac{1}{3}\delta_1, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (48)$$

provided there is no danger of drawing the barrel out of round through the wrist-pin fastenings (see below).

To allow for expansion by heat the inner end of the piston barrel, commencing at the last piston ring groove, is to be tapered down by about from .2 to .5%. The proper amount of taper, that is, the diameter D_1, can only be determined by actual trial, since it differs considerably in different types of engines and with different conditions of operation (for instance, size and design of piston, and kind of iron used, compression pressure, revolutions per minute, heating value of the charge, temperature of cooling water, all have their own effect). Too much play between piston and bore leads to leaking, knocking, and incrustation of the piston; too little increases the friction, hinders lubrication, and causes scoring of the sliding surfaces. Water-cooled trunk pistons are usually only relieved as far as the thickness of the piston face, because the clearance between piston and cylinder bore can be easily regulated by adjusting the temperatures of the cooling water for piston and cylinder jacket. If outside bearings are used for the piston, the clearance between the latter and the cylinder bore is usually made so great that only the piston rings bear against the cylinder wall. (In the piston shown in Figs. 164–166, for instance, the clearance is $51.95''-51.79''=.16''$.)

Dished piston faces and edges projecting internally may cause local accumulation of heat, which is likely to cause trouble for the first piston rings or the wrist-pin bearings. All the rings should if possible be located in that part of the barrel externally cooled (by air or by water), and for that reason it is of advantage to make the bridge up to the first groove (see Fig. 151, p. 141) equal to

$$e \geq 1.0 \text{ to } 1.3\delta. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (49)$$

The thickness of bridge between the grooves, having regard to possible re-turning, should always be $c \geq b$. The depth, t, of the grooves should be but little greater, up to .02″, than the thickness, s, of the rings, especially for the inner rings. The clearance between bottom of groove and the rings should be as small as possible, because it causes loss of charge or of compression, fills with oil, and the oil, crusting down, sticks the rings fast in the grooves.

To protect the piston rings, and to distribute the wear on the cylinder evenly over its entire length, some of the rings are sometimes transferred from the inner hot end, to the outer cool end of the piston. This also tends to retain the oil on the piston surface in better shape. The snap-rings of vertical engines should be secured against turning in their grooves so that the number of joints is evenly distributed over the circumference. In horizontal machines it is usual to space all the joints, with proper staggering, over the lower third of the circumference, or at any rate, care should be taken to avoid any chance of the joints registering with the oil grooves of the cylinder. (See Fig. 153, p. 141.)

To obtain uniform distribution of the bearing pressure K the wrist pin should be situated in the middle of that part of the piston carrying the load; the relieved inner end of the piston should therefore be excluded. Waiving this, the wrist pin is sometimes placed nearer the inner end of the piston in order to obtain a shorter over-all length of machine. Or again, it is sometimes placed nearer the outer end, to obtain a cooler location for the bearing. The latter is compatible with the principles of good design, the former is not.

In large pistons a proper system of webbing has a marked influence on the stiffness and strength. For pistons up to 20″ diameter, from 6–8 radial ribs, each from $\frac{3}{8}''$ to $\frac{5}{8}''$ thick, together with a couple of strong ribs each side of the wrist-pin bosses, is quite sufficient. (See Fig. 160, p. 142.) In larger pistons, at least the radial ribs of the lower half of the piston should be extended to the open end, and the cross-section of the piston through the wrist pin should be stiffened in addition by a couple of circular webs. (See Fig. 155, p. 141.) Thin webs in sufficient number give a lighter weight of piston for equal strength than a few thick webs, and by increasing the radiating surface, render great aid in the cooling of the piston.

Water-cooled trunk pistons receive some stiffening through the added walls of the water space. But care should be taken to see that the wrist-pin bosses are sufficiently stayed with reference to the piston barrel, since no great dependence can be placed in the comparatively weak inner jacket wall. The thickness of this wall is usually kept down in order to decrease weight and to leave room for the head of the connecting-rod, but it should in any case be sufficient to withstand an internal fluid pressure of at least 120–150 lbs. per sq.in. Of course, the pressure of the entering cooling water hardly ever exceeds 60–75 lbs., but there is a possibility of the formation of water hammer, which would cause momentarily considerably higher pressures. If the tightness of the piston face against leakage is to be tested by hydraulic pressure, as is specified for cylinder jackets on p. 128, the test pressure to be used determines also the proper thickness of metal in the jacket wall.

The proper **lubrication** of the piston and its wrist pin is naturally of the utmost importance. Oil grooves on the circumference of the piston are of doubtful value if the oil is fed merely by drop feed and not under pressure. Such grooves are then easily filled up with gummed oil, and become rather harmful than otherwise. If, on the other hand, the oil is forced in by a pump at the right moment, the grooves are washed clean, and are then of service. In the larger engines only forced feed lubrication should be employed, and the oil should be supplied at several points of the piston circumference at the same time. This is the only method which insures certain, and, at the same time, economical lubrication. The oil grooves should not open into the interior of the cylinder. The best place for the exit of the lubricant is between the first and second piston rings (counting from the piston face), or even one ring beyond this. For the lubrication of the wrist pin in small engines some type of drop-feed oiler (supplying the pin by means of a wiper or by a groove along the upper piston surface), or a pressure grease cup, is usually satisfactory; but for the larger sizes only forced lubrication offers sufficient safety, especially in vertical machines.

2. The piston rings are usually cast-iron *snap-rings* of small width, fitted in grooves cut around the circumference of the piston. Special spring rings and detachable seats for the rings have not proved successful, as nearly all these special constructions lose their efficiency after a short time by burning fast or otherwise sticking, and because they usually require a greater thickness of metal in the piston barrel. Rings held out by springs also require a greater depth of grooves, the disadvantage of which has been pointed out above.

In general, the snap-rings are made of uniform thickness. Eccentric rings do not possess any appreciable advantage, because the purpose for which they are used, that is, to obtain a uniform spring around the circumference, is not realized. On the other hand, they cost more to manufacture, and toward the joint the clearance between bottom of groove and the ring is seriously increased.

Constructive Details. Let the external diameter of the snap-ring, when in place, be $D=2r$ ins. (the same as the cylinder bore), let the mean radius of the ring be $r_m=\frac{D-s}{2}$ ins., and let K_b lbs. per sq.in. be the allowable bending stress. Then

$$\frac{r}{s}=\frac{D}{2s}=\sqrt{\frac{K_b}{12p}}+.5,$$

in which p (in pounds per square inch) is the pressure of the ring against the cylinder wall. From this the required *thickness* s of the ring is

$$s=\frac{.5D}{\sqrt{\frac{K_b}{12p}}+.5}\text{ in.} \qquad (50)$$

The allowable stress K_b in the dangerous section opposite the joint, may be taken fairly high, for cast iron from 12000 to 17000 lbs. per sq.in., because the stress is constant and acts always in the same direction. Snap-rings, however, are under their greatest load when being stripped over the piston to slip them into their grooves. This is only possible as long as

$$\frac{r_m}{s}=\frac{D-s}{2s}\geqq\sqrt{\frac{E}{2.5K_b}}, \qquad (51)$$

in which E is the modulus of elasticity.[1]

If it is desired that the stress induced during the slipping on of the rings shall be the same as the stress during operation, that is, that $\sigma'_b=\sigma_b$, we must have a thickness s of ring, assuming $E=11500000$ for cast iron, equal to

$$s=r_m\frac{\sqrt{K_b}}{2145}\text{ in.} \qquad (52)$$

From these equations we find the following relation between $\frac{r}{s}$ and K_b, Table 11.

---

[1] The derivation of these equations is given by K. Reinhardt in the Zeitschrift d. V. D. I., 1901, p. 375. Tables 11 and 12 are taken from this very complete exposition of the question of piston rings.

TABLE 11

PROPORTIONS OF SNAP RINGS TO STRIP OVER PISTON

$\frac{r}{s}=$	20	19	18	17	16	15	14	13	12
$K_b=$	11 600	12 900	14 500	16 350	18 450	20 900	24 300	28 300	33 500 lbs. per sq.in.

The spring pressure, p, per square inch of circumferential area, with which the ring in place presses against the cylinder wall, varies from 3.5 to 7 lbs. per sq.in. Its amount does not appear to depend as much upon the value of the gas pressure to be confined as it does upon the rapidity with which the latter rises and falls.[1] For any given set of dimensions (see Table 12),

$$p=\frac{K_b}{12\frac{r^2{}_m}{s^2}}=\frac{K_b s^2}{12r^2{}_m}\text{ lbs. per sq.in.} \quad . \quad . \quad . \quad . \quad . \quad (53)$$

The *part to be cut out of the ring* at the joint may be made equal to

$$a=9.5\times\frac{D-s}{2}\times\frac{DK_b}{2sE}=9.5r_m\frac{rK_b}{sE}, \quad . \quad . \quad . \quad . \quad . \quad (54)$$

$$a=9.5\times\frac{D-s}{2}\times\kappa=9.5r_m\kappa. \quad . \quad . \quad . \quad . \quad . \quad (54a)$$

Values of $\kappa=\frac{rK_b}{sE}$, the last term of eq. (54), are given in Table 12, for a series of values of $\frac{r}{s}$ and K_b, assuming the value of $E=11500000$ for cast iron.

The *width of the ring* has no influence on the spring p or the bending stress σ_b. It may vary between

$$b=1\tfrac{1}{4}s \text{ and } 2s, \quad . \quad . \quad . \quad . \quad . \quad (55)$$

depending upon the *number of rings* used. The number of rings may be found approximately from

$$i\geqq\frac{D}{5b}, \quad . \quad . \quad . \quad . \quad . \quad (56)$$

in which D=cylinder diameter in inches. It is a matter of experience with gas engines that narrow rings in sufficient number make a tighter packing with less friction than a smaller number of wider rings.

---

[1] A means often resorted to on testing floors, when the piston blows through, is to conduct the pressure of the explosion into the grooves under the rings, to increase the pressure p. Needless to say, this means of "last resort" is wrong from all points of view.

Table 12

VALUES OF p AND κ FOR VARIOUS VALUES OF $\frac{r}{s}$, $\frac{r_m}{s}$, AND K_b, FOR SNAP RINGS

$\frac{r}{s}$	$\frac{r_m}{s}$	$K_b=$	10 000	11 350	12 800	14 200	15 600	17 000	18 450	19 900	21 300
20	19.5	p	2.17	2.49	2.80	3.11	3.42	3.73	4.04	4.36	4.67
		κ	.0175	.0200	.0225	.0250	.0275	.0300	.0326	.0350	.0376
19	18.5	p	2.43	2.77	3.13	3.47	3.81	4.17	4.51	4.87	5.22
		κ	.0166	.0190	.0214	.0238	.0262	.0285	.0308	.0333	.0357
18	17.5	p	2.71	3.10	3.50	3.88	4.27	4.65	5.03	5.43	5.83
		κ	.0157	.0180	.0203	.0225	.0248	.0270	.0293	.0317	.0338
17	16.5	p	3.06	3.48	3.93	4.36	4.80	5.23	5.67	6.11	6.55
		κ	.0149	.0170	.0191	.0212	.0234	.0255	.0276	.0298	.0319
16	15.5	p	3.46	3.96	4.43	4.93	5.43	5.91	6.41	6.91	7.40
		κ	.0140	.0160	.0180	.0200	.0220	.0240	.0260	.0280	.0300
15	14.5	p	3.96	4.51	5.07	5.65	6.22	6.79	7.33	7.90	8.48
		κ	.0131	.0150	.0169	.0188	.0206	.0225	.0244	.0262	.0282
14	13.5	p	4.56	5.21	5.99	6.53	7.16	7.89	8.47	9.11	9.75
		κ	.0122	.0140	.0157	.0175	.0192	.0210	.0228	.0245	.0263
13	12.5	p	5.32	6.07	6.83	7.60	8.34	9.10	9.86	10.62	11.35
		κ	.0114	.0130	.0146	.0163	.0179	.0195	.0212	.0228	.0244
12	11.5	p	6.27	7.17	8.10	8.97	9.88	10.76	11.65	12.55	13.42
		κ	.0105	.0120	.0135	.0150	.0165	.0180	.0195	.0210	.0225
11	10.5	p	7.53	8.61	9.66	10.72	11.80	12.85	13.95	15.05	16.10
		κ	.0096	.0110	.0124	.0138	.0151	.0165	.0179	.0193	.0207
10	9.5	p	9.18	10.50	11.80	13.12	14.40	15.60	17.00	18.35	19.70
		κ	.0087	.0100	.0113	.0125	.0138	.0150	.0163	.0175	.0188
9	8.5	p	11.50	13.12	14.75	16.40	18.00	19.70	21.30	23.00	24.56
		κ	.0079	.0090	.0101	.0112	.0124	.0135	.0146	.0157	.0169

Examples. 1. A piston whose diameter is $D=35.5''$ is to have snap rings, which, when stripped over the piston, shall not have a stress in outer fiber exceeding $K_b=18450$ lbs. per sq.in.

From Table 11, when $K_b=18450$, the ratio $\frac{r}{s}=16$.

From Table 12, for $K_b=18450$ and $\frac{r}{s}=16$, p, the pressure exerted by the ring, $=6.41$ lbs. per sq.in. Hence the thickness of ring will be

$$s=\frac{.5D}{\sqrt{\frac{K_b}{12p}+.5}}=\frac{.5\times 35.5}{\sqrt{\frac{18450}{76.9}+.5}}=1.10''.$$

If the width b is chosen $=1.25\ s=1.375''$, the number of rings required will be

$$i=\frac{35.5}{5\times 1.375}=5 \text{ or } 6.$$

From Table 12 we also find the value of κ to determine the part of ring to be cut out at the joint. From eq. (54a) we have

$$a=9.5\times\frac{D-s}{2}\times\kappa=9.5\times\frac{35.5-1.10}{2}\times .026=4.25''$$

2. Suppose that, for the same piston, the specifications call for a value of $p=5.7$ lbs. per sq.in. and a value of $K_b=14200$ lbs. when the ring is in service.

Then from Table 12 the value of $\frac{r}{s}$ will be about 15, that is, the thickness of the ring will be

$$s=\frac{r}{15}=\frac{35.5}{2\times 15}=1.18''.$$

The width of ring is $1.25\times 1.18=1.475''$, and the required number is $i=\frac{35.5}{5\times 1.475}=5$. The part cut out of the circumference will be, again using κ from Table 12,

$$a=9.5\times\frac{35.5-1.18}{2}\times .0188=3.06''.$$

If it is desired to strip these rings in place, the value of $\frac{r_m}{s}$ should be, from eq. (51), $=\sqrt{\frac{11500000}{2.5\times 14200}}=18.0$. But the above dimensions show a value of $\frac{r_m}{s}=\frac{35.5-1.18}{2\times 1.18}=14.5$; hence stripping is no longer possible unless there is no objection to K_b exceeding 14200 lbs. per sq.in. Another way of solving the difficulty, besides furnishing detachable fastenings for these rings, would be to use a cast iron of greater ductility, that is, smaller value of E.

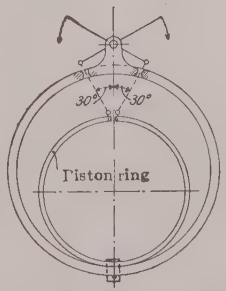

FIG. 172. — Spreader used to put snap ring in place.

The spreading of the rings preparatory to stripping them into place must be carefully done, otherwise even a properly made ring may break. With rings of large diameter, spreading is possible only by means of special appliances. Fig. 172 shows one of the types proposed by Reinhardt. This is so designed that the angle between the two forces tending to spread the ring is always about 60°. The greater this angle, the greater the danger of fracture, a fact which should be kept in mind when spreading by hand.

To determine the **shop dimensions** for the ring, express the gap distance a, see Fig. 173 b, as a function of the cylinder diameter, say $a=xD$, or $x=\frac{a}{D}$. Then for the *preliminary turning* make

the external diameter $$D_a=\left[D\left(1+\frac{x}{\pi}\right)+.12\right] \text{ in.,} \qquad (57)$$

the internal diameter, $$D_i=\left[D\left(1+\frac{x}{\pi}\right)-2s-.12\right] \text{ in.} \qquad (58)$$

After cutting out the gap a [a determined from eq. (54) and (54a)], the external diameter of the compressed ring will then be

$$D_0=\frac{D_a\pi-a}{\pi}=(D+.12) \text{ in.} \qquad (59)$$

The mean allowance for finishing is therefore .12''. This factor disappears in eq. (58) if the rings are not to be finished on the inside. Large piston rings require a somewhat greater allowance, while for small rings a somewhat smaller allowance than .12'' is sufficient.

Example. 3. For the rings computed in Example 1, the gap $a=4.25''$, hence $x=4.25\div35.5=.118$. Then

$$D_a=35.5\left(1+\frac{.118}{\pi}\right)+.12=36.96'';$$

$$D_i=35.5\left(1+\frac{.118}{\pi}\right)-(2\times1.10)-.12=34.52'';$$

$$D_0=\frac{36.96\pi-4.25}{\pi}=35.62''=(D+.12)\text{ in., as called for.}$$

Notes regarding the **manufacture** of piston rings: The rings are turned and cut off from a piece of cast-iron pipe having sufficient wall thickness. Stress should be laid on obtaining the proper grade of iron, and it is a good idea to have the pipe cast solid at one end.

The work proceeds about as follows: (see Fig. 173)

(*a*) Turn the pipe to the proper diameter, inside and out, leaving a certain allowance (see above) to give the finished ring the required spring when compressed.

(*b*) Cut off the rings at once to the required width, so that possibly only slight scraping will be necessary to make them fit the grooves closely.

(*c*) Cut out of the circumference a certain distance or gap *a*, see Fig. 173*b*, so that the ring may be compressed to the diameter D_0. For the ordinary lap joint, the piece *a* may be cut out by milling (Fig. 173*a* above), or by drilling (Fig. 173*a* below). Compressing the ring to the diameter D_0, however, gives it a slightly elliptical form, which, in the case of small rings, may be taken care of in the allowance made for finishing. In large rings, however, this eccentricity must be removed by hammering the inner surface of the ring. This procedure at the same time increases the spring of the ring. The surfaces of the joint in contact should be scraped to a fit.

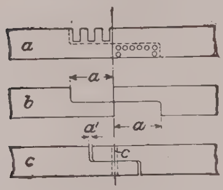

FIG. 173.

(*d*) Join the ends of the ring. This may be done as in Fig. 173*c*, by passing a pin *c* through the two ends held the proper distance apart, or if the joint is of the simple diagonal type, two pieces of thin sheet steel may be riveted over the cut. In both cases the method of fastening should be somewhat flexible to prevent any straining or warping of the ring.

(*e*) The rings are next turned to the desired outside diameter, and also inside diameter, if required. This may be done by turning them one at a time or by holding a number of them in a suitable jig. For very important work the rings should not be finished at one cut or at one setting; instead, a number of smaller cuts should be taken and the hold loosed after every cut to allow of equalization of any possible casting stresses. Rings which must be very accurate (for high pressure work) should also be finished one at a time.

(*f*) After the removal of the joint connection, file out the clearance *a'*, Fig. 173*c*, to an amount varying from .04 to .20", depending upon the diameter of the ring.

The step joint, Fig. 173, is only used in rings which have a diameter exceeding 8" and a width of at least ½". For smaller rings the joint is simply straight diagonal or straight across.

The **diameter of the wrist pin,** sometimes called piston pin, depends upon the moment

$$M_b=\frac{P_z l_0}{4}=p_z\times .785\ D^2\times\frac{l_0}{4}\ \text{in.-lbs.,}\quad\ldots\ldots\ldots\ldots\quad(60)$$

and

$$M_b\leqq\tfrac{1}{10}d^3K_b\ \text{in.-lbs.,}$$

where d is the diameter of the pin, for l_0 see Fig. 154, p. 141.

For given values of d, l_0, and P_z, the bending stress in the pin therefore is

$$\sigma_b=\frac{10\ M_b}{d^3}=\frac{10\times p_z\times .785\ D^2 l_0}{4d^3}=\frac{1.964 p_z\ D^2 l_0}{d^3}\ \text{lbs. per sq.in.,}\quad\ldots\quad(61)$$

and from this, putting $K_b=12000$ lbs. per sq.in. for steel, the least diameter of the pin must be

$$d=\sqrt[3]{\frac{1.964 p_z\ D^2 l_0}{K_b}}=\sqrt[3]{\frac{p_z\ D^2 l_0}{6000}}\ \text{in.}\quad\ldots\ldots\ldots\ldots\quad(62)$$

For our normal diagram, with $p_z=356$ lbs.,

$$d=\sqrt[3]{\frac{D^2 l_0}{17}}\ \text{in.}\quad\ldots\ldots\ldots\ldots\quad(62n)$$

Eqs. (60) to (62) depend upon the assumption that the pin is supported in its bearings with some flexibility and that the load is concentrated at the middle of the pin. In fact, the load distributes itself over a length of the pin, which in the most favorable case may reach the entire bearing length l. In such cases the bending moment reduces to

$$M_b=\frac{P_z}{2}\left(\frac{l_0}{2}-\frac{l}{4}\right)\ \text{in.-lbs.}\quad\ldots\ldots\ldots\ldots\quad(63)$$

Bach, for instance, uses this formula for cross-head pins. In view of the fact that, in new machines, the pin hardly ever bears over its entire length, and that any deformations of piston or pin bearing will destroy this condition, if it ever exists, eqs. (60) to (62) should have the preference.

The length of the pin depends upon the unit bearing pressure, which, at the moment of explosion, should not exceed 1800 lbs. per sq.in., and should be smaller if possible. Since

$$K_{\max}=\frac{P_z}{dl}=\frac{p_z\times .785\ D^2}{dl}\ \text{lbs.,}\quad\ldots\ldots\ldots\ldots\quad(64)$$

the length of the pin should be

$$l\geqq\frac{P_z}{dK_{\max}}=\frac{p_z\times .785\ D^2}{dK_{\max}}\ \text{in.,}\quad\ldots\ldots\ldots\ldots\quad(65)$$

and with $p_z=356$ lbs. per sq.in., and $K_{\max}=1800$ lbs. per sq.in.,

$$l\geqq\frac{.16\ D^2}{d}\ \text{in.}\quad\ldots\ldots\ldots\ldots\quad(65n)$$

In large engines with high explosion pressure, the room available for the inner connecting-rod end is in nearly all cases so limited as to necessitate the use of unit-bearing pressure greater than the above (up to 21–2200 lbs. per sq.in.). Under such conditions, however, the wrist pin, even if carefully constructed and well lubricated, will always be the most delicate part of the entire machine. In every case the pin should be made of the best machinery steel and the bearing surfaces hardened.

In small engines the pin is held in its bearings, which are nearly always slightly tapered, by strong set-screws, and is kept from turning by keys. (See Figs. 151–154, p. 141.) For large engines, of course, more elaborate methods than this are required, some of which are shown in Figs. 155–161. Although they should insure the greatest safety, the pin fastenings should be such as to be easily loosened, and when drawn up they should not tend to deform the piston. This should be especially noted in key fastenings.

The dimensions of **piston rods** may be determined from the generally applicable long column formula of Euler, which may also be used for connecting-rods, as will be explained more in detail later. The material of construction for piston rods should be the best grade of ductile open-hearth or crucible steel.

Formulæ (1) to (3) or (3*n*), p. 191, may be applied without modification to solid piston rods. The length L is figured from the center of the cross-head pin to the middle of the length of the piston boss. The factor of safety $\mathfrak{S}=20$, used there for other reasons, should not be materially reduced for piston rods since the conditions of their operation are no more favorable, and any subsequent returning of the rods on account of wear must be allowed for. Ample rod diameters are also desirable with reference to stuffing-box construction (see p. 138.)

Usually the piston rods of internal-combustion engines are hollow to allow of the passage of cooling water. The moment of inertia of the rod cross-section is then no longer

$$I=\frac{\pi d^4}{64}, \quad \text{but} \quad I=\frac{\pi}{64}(d^4-d_i^4)\sim\tfrac{1}{20}(d^4-d_i^4) \text{ in.}^4, \quad . \quad . \quad . \quad . \quad . \quad (66)$$

where d and d_i stand respectively for the external and internal diameters of the rod.

From the general long column formula, the allowable load on a column is

$$P_k=\pi^2\frac{EI}{\mathfrak{S}L^2} \text{ lbs.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (67)$$

For $E=30000000$ and $\mathfrak{S}=20$, the allowable total piston pressure then becomes

$$P_z=9.87\frac{30000000}{20^2L^2}(d^4-d_i^4)\sim\frac{740000(d^4-d_i^4)}{L^2} \text{ lbs.} \quad . \quad . \quad . \quad . \quad . \quad . \quad (68)$$

For our assumed normal pressure $=356$ lbs. per sq.in., $P_z=280\ D^2$, hence

$$D^2\leqq\frac{2640(d^4-d_i^4)}{L^2}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (68n)$$

In vertical engines, eqs. (68) or (68*n*) may be directly applied. With the factor of safety chosen, $\mathfrak{S}=20$, the resulting moment of inertia is generally so large that a correction with reference to the additional bending moment, due to weight of rod and piston in horizontal machines, is not necessary. To be entirely certain, however, it is well to check the deflection, f, according to the formula

$$f=\frac{L^3}{48}\,\frac{\frac{5}{8}G_s+G_k}{EI}\text{ in. for machines with tail-rod,} \quad \ldots \ldots \quad (69)$$

and

$$f=\frac{L^3}{EI}\left[\frac{G_s}{8}+\frac{G_k}{3}\right]\text{ in. for machines without tail-rod,} \quad \ldots \quad (69a)$$

in which G_s is the weight of piston rod, and G_k is the weight of piston including its charge of cooling water. f must be kept in any case smaller than the clearance allowed between piston and cylinder bore.

The deflection of the piston rod should be confined to its lowest possible amount on account of its effect on the stuffing-boxes. If with the maximum deflection allowable, eq. (69) demands too great rod diameters, either the piston must be allowed to rest on the bottom of the cylinder, or some such scheme as Kollman's method of centering must be adopted. This consists in giving the rod, when under no load, a bend upward during the manufacture equal to the deflection f, so that, when in service, the weights G_s and G_k draw the rod into a straight line. In the shop the rod is given this initial bend by the use of a revolving tool head, holding the rod stationary. It cannot be done by the ordinary method of turning between centers.

For information concerning the methods of fastening the piston on the rod, see pages 141 and 142.

V. Crank Shafts

Material. Cast steel, for the larger engines only the very best grades; mild crucible steel with a tensile strength of at least 70000 lbs. per sq.in., and a ductility of at least 20%, or open-hearth steel with a tensile strength of at least 64000 lbs. per sq.in. and 18–20% elongation. Of late years, in very important cases, a high percentage nickel steel is sometimes used. This has a tensile strength of from 96000–98000 lbs. per sq.in with an elongation of from 20–22% (in 8″).

Allowable stress about 12000 lbs. per sq.in.; with excellent material and well defined stress conditions this may be increased to 14000 lbs. per sq.in. In crank shafts, however, besides the question of strength, the stiffness, that is, safety against bending, is the all-important consideration, and the dimensions should be taken accordingly.

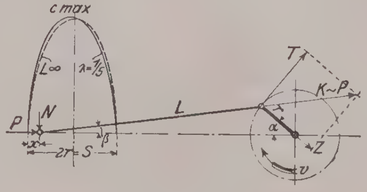

FIG. 174.

1. General Kinematic and Stress Relations. Referring to the notation of Fig. 174, the piston position, or portion of the stroke passed over corresponding to any given crank angle α, may be expressed by

$$x=r(1-\cos\alpha)\pm L(1-\cos\beta)\sim r(1-\cos\alpha)\pm\frac{1}{2}\frac{r^2}{L}\sin^2\alpha, \quad \ldots \ldots \quad (1)$$

or, with a ratio of crank to connecting-rod $\lambda=\frac{r}{L}$,

$$x=r(1-\cos\alpha)\pm L[1-\sqrt{1-(\lambda\sin\alpha)^2}]\sim r(1-\cos\alpha\pm\tfrac{1}{2}\lambda\sin^2\alpha). \quad \ldots \quad (1a)$$

In these equations, and in what follows, the positive sign will refer to the outstroke, and the negative sign to the instroke of the piston.

In crank mechanisms in which the center line of the cylinder does not pass through the center of the crank-shaft, the stroke of the piston is greater than the diameter of the crank circle. If the offset between cylinder and center of shaft $=a$, the stroke of the piston will be

$$S=\sqrt{(L+r)^2-a^2}-\sqrt{(L-r)^2-a^2}. \quad \dots \quad (2)$$

The increase or decrease of the *piston velocity* may be found from the expression

$$c=v\frac{\sin(\alpha\pm\beta)}{\cos\beta}\sim v\sin\alpha(1\pm\lambda\cos\alpha), \quad \dots \quad (3)$$

in which $v=\frac{2r\pi n}{60}=\frac{r\pi n}{30}=$uniform linear velocity of the crank pin.

The acceleration, or retardation, of the piston from one velocity c_1 to another velocity c_2, in the time interval from t_1 to t_2, expressed in seconds, may be expressed by

$$b=\frac{c_2-c_1}{t_2-t_1}\sim\frac{v^2}{r}(\cos\alpha\pm\lambda\cos 2\alpha)=r\omega^2(\cos\alpha\pm\lambda\cos 2\alpha). \quad \dots \quad (4)$$

The piston has its maximum velocity when $\alpha+\beta=90°$, at which position

$$\tan\beta=\lambda=\frac{r}{L}, \quad \text{and} \quad c_{\max}=\frac{v}{\cos\beta}. \quad \dots \quad (5)$$

Assuming $\lambda=\frac{1}{5}$, we shall then have

$$c_{\max}=1.02v=1.6c_m, \quad \dots \quad (5a)$$

in which c_m, the *mean piston velocity* $=\frac{nS}{30}$. . . . . (6)

The force of acceleration, or the inertia pressure, of the reciprocating parts of an engine per square inch of piston, is, when the weight of these parts per square inch of piston is expressed by $G_0=\frac{G_t}{F}$,

$$p_b=\frac{G_0}{g}b=r\omega^2\frac{G_0}{g}(\cos\alpha\pm\lambda\cos 2\alpha). \quad \dots \quad (7)$$

Since the angular velocity of the crank $\omega=\frac{n\pi}{30}$, we may write eq. (7),

$$p_b=r\pi^2\frac{G_0}{g}\left(\frac{n}{30}\right)^2(\cos\alpha\pm\lambda\cos 2\alpha)\sim .306\left(\frac{n}{30}\right)^2G_0r(\cos\alpha\pm\lambda\cos 2\alpha). \quad \dots \quad (7a)$$

From eq. (7*a*) we have for the two dead center positions of the piston

$$p_{b_1}=.306\left(\frac{n}{30}\right)^2G_0r(1\pm\lambda)=.306\left(\frac{n}{30}\right)^2G_0r\left(1\pm\frac{r}{L}\right). \quad \dots \quad (7b)$$

The total weight, G_t, is made up mainly of the reciprocating parts (piston, cross-head, and piston rod). Only part of the weight of the connecting-rod (from .45 to .55 of its total weight) is included in the weight G_t or G_0. The accurate determination of the inertia effect of the connecting-rod is best made graphically.[1]

TABLE 13

WEIGHTS OF RECIPROCATING PARTS PER SQUARE INCH OF PISTON FOR VARIOUS TYPES OF ENGINES

Kind of Engine.	Weight $G_0=\frac{G_t}{F}$ lbs. per in. of Piston.
Single-acting Engines:	
Trunk piston, $S \leqq 1.75\ D$	3.5–4.0
Trunk piston, $S \geqq 2.0\ D$	4.2–4.9
With cross-head, $S=1.5$ to $1.75\ D$	5.6–7.0
Small high-speed automobile engines	.35–.60
Double-acting 4-cycle Engines:	
Single cylinder	11.0–17.5
Tandem	18.0–25.0
Double-acting 2-cycle Engines:	
Small (300 H.P. Körting)	16.0
Large (1000 H.P. Simplex)	21.4

If p_i represents the fluid (gas) pressure behind the piston at any time, the net pressure on the piston during the time the reciprocating parts are accelerated will be p_i-p_b, and during the time of retardation it will be p_i+p_b lbs. per sq.in. But since the work of acceleration during one stroke of the piston must equal the work of retardation, i.e., the sum of these is equal to zero, the power of the engine is independent of p_b and only depends upon p_i. In moderate speed stationary engines the inertia pressure, p_b, will not usually exceed from 40–60 lbs. per sq.in. of piston; but in high-speed motors it may under certain circumstances be much more than this. Fig. 175 shows well what abnormal operating conditions may occur with reference to the inertia pressures, when the weight of the reciprocating parts is not well adapted to the speeds used. The figures refer to an automobile engine with $D=3\frac{3}{8}''$, $S=5\frac{1}{2}''$, $G_0=.64$ lbs. per sq.in. of piston, and $\lambda=\frac{1}{4.5}$.

The total pressure on the piston at any given time is

$$P=p_i\times .785\ D^2.$$

This causes a normal pressure upon the guide (trunk piston or cross-head) equal to (see Fig. 174),

$$N=P\tan\beta. \qquad (8)$$

The pressure along the rod is

$$K=\frac{P}{\cos\beta}.$$

---

[1] Mollier's Method, described in Zeitschrift d. V. D. I., is well suited to this purpose.

The difference between P and K is so small with the connecting-rod ratios in common use, that they may in general be considered equal.

The turning force (tangential crank-pin pressure) is

$$T=P\frac{\sin(\alpha+\beta)}{\cos\beta}, \quad \ldots \ldots \ldots \ldots \quad (9)$$

and, finally, the bearing pressure is

$$Z=\frac{P}{\cos\beta}\cos(\alpha+\beta)\sim P\cos(\alpha+\beta). \quad \ldots \ldots \ldots \quad (10)$$

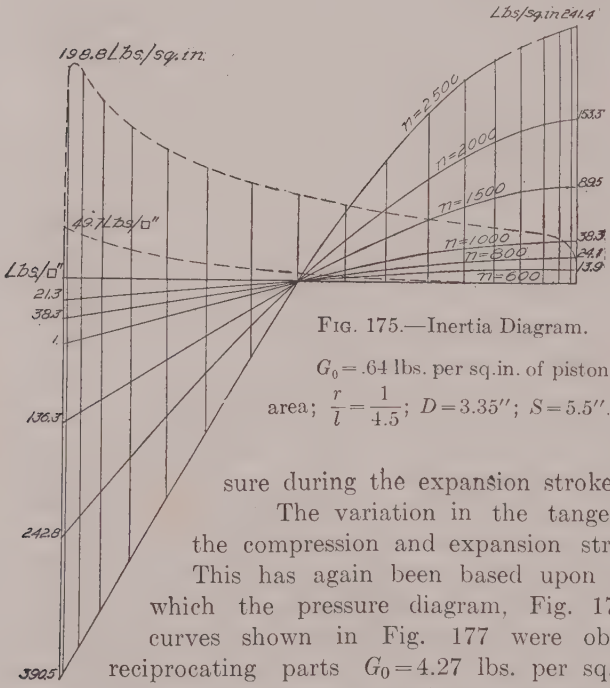

FIG. 175.—Inertia Diagram.

$G_0=.64$ lbs. per sq.in. of piston area; $\frac{r}{l}=\frac{1}{4.5}$; $D=3.35''$; $S=5.5''$.

The curve, Fig. 176, shows the variation in the value of the guide pressure, N, for the expansion stroke, as computed on the basis of the normal diagram, Fig. 33, p. 71, and for the ratio $\lambda=\frac{1}{5}$. For the other three strokes N is of course much smaller. Since the pressure volume diagrams for all explosion engines have about the same general shape, Fig. 176 shows the general variation in N. As will be seen, $N_{\max}$ occurs between 16 and 18% of the stroke, and at that point has the value

$$N_{\max}\sim\tfrac{1}{10}P_z. \quad . \quad . \quad (11n)$$

This figure is rounded off, taking into account any possible late ignition. The mean guide pressure during the expansion stroke is about $N_m\sim\frac{1}{3}N_{\max}=\frac{1}{30}P_z$.

The variation in the tangential effort or turning force during the compression and expansion strokes may be seen from Fig. 178. This has again been based upon the normal diagram, Fig. 33, from which the pressure diagram, Fig. 177, has been derived. The inertia curves shown in Fig. 177 were obtained by assuming the weight of reciprocating parts $G_0=4.27$ lbs. per sq.in. of piston and a speed of 200 r.p.m. The inertia forces for the suction and expansion strokes, during which they are not taken up by gas pressure, cause an alternately positive and negative pressure on the crank-pin, which pressure is considerably greater than that due to the suction and exhaust resistances. Hence the effect of these resistances may be neglected and the tangential effort diagram confined to the second and third strokes of the cycle. Figs. 177 and 178 apply to 2-cycle engines as well as to 4-cycle. Complete tangential effort diagrams are given under "Computation of Fly-wheels," p. 215.

From Fig. 178 it appears that $T_{\max}$ occurs at approximately 40° crank angle. The factor $\frac{\sin(\alpha+\beta)}{\cos\beta}$ in eq. (9) for this crank angle ($\alpha=40°$) is approximately $=.75$. The pressure still remaining on the piston for the same angle is usually in the neighborhood of $P_t \sim .6\ P_z$. Hence the maximum tangential pressure may be expressed by

$$T_{\max} = .75\ P_t = .75 \times .6\ P_z = .45\ P_z \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (12n)$$

Without correction for inertia, the value of P_t and hence of $T_{\max}$, would be somewhat greater, and since when an engine is being started, the inertia forces are comparatively insignificant, it will be better for the sake of safety to write in general

$$\mathbf{P}_t = \mathbf{.7\ P}_z, \quad . \quad . \quad . \quad (13n) \qquad \text{and} \qquad \mathbf{T}_{\max} = \mathbf{.5\ P}_z. \quad . \quad . \quad . \quad . \quad . \quad (14n)$$

The above also makes some allowance for possible late ignitions.

The shaft bearings are always under the maximum pressure at the moment of explosion. Hence if the two main bearings are placed symmetrically with reference to the center line of the cylinder, the maximum pressure on each will be $Z = \frac{P_z}{2}$ lbs.

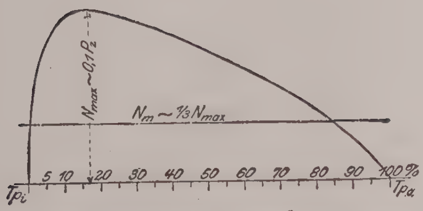

FIG. 176.—Variation in Pressure on Guide.

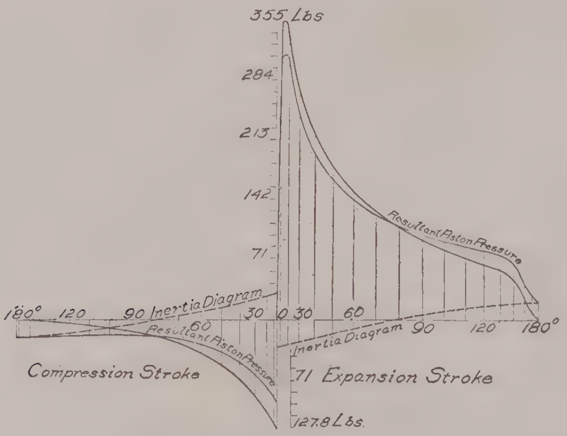

FIG. 177.—Pressure Diagram.

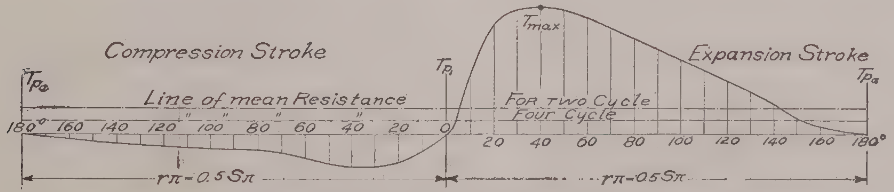

FIG. 178.—Forming Effort Diagram for Compression and Expansion Strokes.

The magnitude and direction of Z is, however, modified by the weight of shaft and of fly-wheel, and by the belt pull, if this method of transmitting the power is used.

Table 14 gives the complex trigonometric functions used in computing piston positions, velocities, pressures, etc., for various per cents of the out- and instroke.

TABLE 14

PISTON AND CRANK TRAVEL, WITH THE TRIGONOMETRIC FUNCTIONS INVOLVED

1	2	3	4	5	6	7	8
Piston Position x.		Crank Angle * Outstroke α	Rod Angle, β	$\sin(\alpha+\beta)$	$\cos(\alpha+\beta)$	$\sin\alpha\times(1+\frac{1}{5}\cos\alpha)$	$\cos\alpha\pm\frac{1}{5}\cos 2\alpha$
↓ Outstroke in % S.	↑ Instroke in % S.						
2	98	14.8	3.0	0.306	0.952	0.304	1.141
4	96	21.2	4.1	0.425	0.905	0.428	1.080
6	94	26.0	5.0	0.515	0.857	0.516	1.022
8	92	30.2	5.4	0.583	0.812	0.586	0.955
10	90	33.9	6.2	0.630	0.775	0.648	0.905
12	88	37.3	7.0	0.698	0.715	0.705	0.848
15	85	41.8	7.4	0.757	0.654	0.765	0.767
20	80	48.9	8.4	0.842	0.540	0.853	0.631
25	75	55.4	9.3	0.904	0.428	0.914	0.498
30	70	61.5	10.1	0.949	0.315	0.962	0.369
35	65	67.3	10.4	0.977	0.214	0.993	0.244
40	60	73.0	11.0	0.994	0.105	1.011	0.126
45	55	78.6	11.2	1.000	0.003	1.018	0.012
50	50	84.2	11.3	0.995	0.096	1.014	0.068
55	45	90.0	11.4	0.980	0.196	1.000	0.200
60	40	95.7	11.3	0.956	0.292	0.976	0.295
65	35	101.7	11.2	0.920	0.390	0.939	0.388
70	30	108.0	11.0	0.875	0.485	0.892	0.471
75	25	114.6	10.3	0.839	0.545	0.832	0.548
80	20	121.9	9.5	0.751	0.660	0.759	0.616
85	15	130.0	8.6	0.663	0.749	0.668	0.677
88	12	135.3	8.0	0.597	0.802	0.604	0.709
90	10	139.2	7.3	0.552	0.834	0.554	0.727
92	8	143.6	6.5	0.500	0.866	0.496	0.746
94	6	148.5	6.0	0.431	0.903	0.433	0.755
96	4	154.3	5.0	0.353	0.936	0.355	0.776
98	2	161.9	3.4	0.253	0.967	0.251	0.789
100	0	180.0	0	0	1.000	1.000	0.800

* Crank angle for instroke $=180-\alpha$.

2. Friction Losses in Journals. The friction losses considered in the design of bearings and pins should be computed on the basis of the *average* piston or journal pressure for one *complete* cycle (i.e., for four or two strokes, as the case may be). On account of the extreme variations in the working pressures and the uncertainty regarding the coefficient of friction existing at any given instant, the allowable pressures on the sliding surfaces in contact should be taken at a rather low figure. The explosion pressure, p_z, only determines the *maximum* unit pressure $k_{\max.}$ on the journal, and this should be placed at such a figure that the oil film can never be completely squeezed out of the bearing. This is satisfied in general as long as $k_{\max.} \leqq 1400$ to 1600 lbs. per sq.in.; higher values of k up to 1800 lbs. per sq.in. are permissible only in extreme cases.

In the case of 4-cycle engines the average piston pressure of a complete cycle is composed of the average pressures on the four strokes, that is,

$$p_m=\frac{p_1+p_2+p_3+p_4}{4} \text{ lbs. per sq.in.}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (15)$$

p_1 and p_4 for the suction and exhaust strokes are due to the inertia forces which the crank-pin must take up. For the mean pressures of the compression and expansion stroke, i.e., p_2 and p_3, this effect disappears, since the positive pressures due to the moving parts during the one half of the strokes are balanced by the negative pressures during the other half. Turning to our standard diagram, Fig. 33, p. 71, we find p_1 and p_4 each equal to about 18 lbs. per sq.in., $p_2=32$ lbs. per sq.in., and $p_3=128$ lbs. per sq.in. $(p_3-p_2=p_i)$. Hence

$$p_m=\frac{18+32+128+18}{4}=49 \text{ lbs. per sq.in.} \sim 50 \text{ lbs. per sq.in.} \quad . \quad . \quad . \quad (16n)$$

With a given diameter of cylinder equal to D in., the resulting mean piston pressure for one complete cycle therefore is

$$P_m=50\times .785\ D^2=39.25\ D^2\sim 40\ D^2 \text{ lbs.} \quad . \quad . \quad . \quad (17n)$$

It is quite usual even to-day to compute the work lost in friction using the mean effective pressure, p_i, or even the explosion pressure p_z. Either method is incorrect, and the computation so made can at best only give some general idea of the magnitude of the friction loss for preliminary estimates. This method of computation would mean that 4-cycle and 2-cycle engines, single- and double-acting engines, should all have the same crank-pin dimensions, which of course is wrong.

The general equation for the work lost in friction is

$$A_r=kv\mu=\frac{P}{dl}\times\frac{d\pi n\mu}{60\times 12} \text{ ft.-lbs. per sec.} \quad . \quad . \quad . \quad (18)$$

Equation (18) contains the coefficient of friction, μ, which, depending upon the load, quality of the surfaces in contact, and the degree of lubrication, may vary from .03 to .10. If, on account of its uncertainly, this factor is entirely neglected (it cancels from both sides of the equation), we will have the following general formula:

$$kv=\frac{P\pi n}{720l}\sim\frac{Pn}{230l} \text{ ft.-lbs. per sec.} \quad . \quad . \quad . \quad (19)$$

For the crank pin we therefore have

$$kv=\frac{p_m\times .785\ D^2n}{230l}=\frac{p_mD^2n}{290l} \text{ ft.-lbs. per sec.} \quad . \quad . \quad . \quad (19a)$$

From the value for p_m computed above for the standard diagram, we may further derive the special equation

$$kv=\frac{50\ D^2n}{290l}=\frac{D^2n}{5.8l} \text{ ft.-lbs. per sec.} \quad . \quad . \quad . \quad (20n)$$

(*a*) **The Crank Pin.** *Without reference to considerations of strength,* the minimum crank-pin length should therefore be from eq. (19*a*)

$$l=\frac{p_mD^2n}{290kv} \text{ in.,} \quad . \quad . \quad . \quad (21)$$

or, for the standard diagram, p. 71, we have according to eq. (20n),

$$l=\frac{D^2n}{5.8kv}\text{ in.} \quad . \ . \ . \ . \ . \ . \ . \ . \ . \ . \ . \ . \ . \quad (21n)$$

For the ordinary construction of steel crank pin, the maximum limiting values of kv, as based on experience, are about as follows:

For Bronze Bearings.	For Bearings lined with White Metal.
$kv \leqq 1200$ ft.-lbs. per sec.,	$kv \leqq 1500$ ft.-lbs. per sec.

Where very careful construction and perfect lubrication is assured, the value of kv for white metal can in case of necessity be increased to $kv=1650$ ft.-lbs. per sec. It is recommended, however, that this value be used in extreme cases only.

In order to retain the oil film in the bearing during the moment of explosion, we must have

$$k_{\max}=\frac{p_z\times .785\ D^2}{dl}\leqq 1400\text{ to }1700\text{ lbs. per sq.in.,} \quad . \ . \ . \ . \ . \quad (22)$$

or

$$k_{\max}=\frac{280\ D^2}{dl}\leqq 1400\text{ to }1700\text{ lbs. per sq.in.} \quad . \ . \ . \ . \ . \ . \ . \ . \quad (22n)$$

For this reason we should be certain that

$$l\geqq\frac{p_zD^2}{2000d}\text{ in.,} \quad . \ . \ . \ . \quad (23) \qquad \text{or} \qquad l\geqq\frac{D^2}{5d}\text{ in.} \quad . \ . \ . \ . \ . \quad (23n)$$

In the case of small and medium sized engines, other features of design, especially the crank-pin bearing of the connecting-rod, will generally result in dimensions for which kv and $k_{\max}$ are smaller, i.e., l is larger than the values above given.

(*b*) **Crank Shaft or Main Journals.** The bearing pressure in the main journals is determined by the piston pressure, the weight of the fly-wheel, and the pull of the belt, neglecting the weight of the shaft itself. The weight of the wheel and the belt pull generally affect only one of the bearings directly, but the general practice is to neglect this fact also. In the case of vertical engines with shaft above the cylinder, the bearing away from the wheel is relieved of a part of the explosion load, P_z, by the weight of the wheel, G, because weight of wheel and force of explosion are opposite in direction (most favorable case). In vertical engines of ordinary design, with crank shaft below the cylinder, the directions of P_z and G are the same, hence these forces are additive, and the bearing next the wheel receives an extra load due to G (most unfavorable case). In horizontal engines, the line of action of G is at right angles to that of P_z, hence the resulting bearing pressures are relatively less than in vertical engines with crank shaft below the cylinder. However, the line of action of the bearing pressure due to the gas pressure in the cylinder approaches a vertical more and more as the expansion proceeds, and this effect may be emphasized under certain conditions by the belt pull. For that reason, the following investigation, based on the assumption that the lines of action of P_z and G coincide, may be applied to all types of construction.

Referring to Fig. 179, we will then have

$$kv=\frac{.5\,P_m+G\dfrac{a+a_1}{a}}{d_1l_1}\times\frac{\pi d_1n}{60\times12}=\frac{.5\,P_m+G\dfrac{a+a_1}{a}}{l_1}\times\frac{n}{230}\leqq 700\text{ ft.-lbs. per sec.,}\quad .\quad (24)$$

from which the required minimum length of main journal may be determined.

For engines having a third, or outboard bearing, we have the following relation for the second main bearing:

$$kv=\frac{.5\,P_m+G\dfrac{a_2}{a_1+a_2}}{l_1}\times\frac{n}{230}\leqq 700\text{ ft.-lbs.per sec.}\ .\ .\ .\ .\ .\ .\ .\quad (25)$$

In the case of main journals, as in the case of crank pins, the above limiting value for kv, which applies to ordinary constructions, may under favorable circumstances be exceeded, and to a considerable extent, as some of the examples to follow will show.

The weight of any belt wheel is usually so small as compared with the weight of the fly-wheel, that only the latter need be considered. In any exceptional case these two gravity forces must be combined. The maximum bearing pressure in the outboard bearing is usually kept below 400 lbs. per sq.in.

3. Strength Computations. Side cranks are used but little in gas-engine construction, except perhaps for some small and cheap engines for general power purposes. The objection to them is mainly based on the fact that these cranks are supported on one side only.[1]

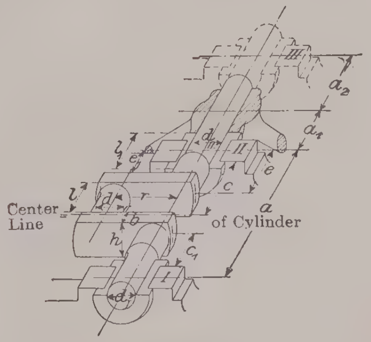

FIG. 179.

The usual type of crank-shaft (center crank) is given a third, or outboard, bearing when the fly-wheel is exceptionally heavy, or when it is seated any considerable distance from the main bearing. Above 30 H.P. the use of an outboard bearing is more common. In nearly every case the center line of the cylinder bisects the distance between the middle planes of the main bearings, i.e., referring to Fig. 179, the distance on each side is equal to $\frac{1}{2}a$. The usual proportions are

$$a=\text{from }1.8\text{ to }2.2\,D,$$

$$a_1=\text{from }.4\text{ to }.5a,$$

and the center line of the third bearing is usually at a distance $a_1+a_2=$from 1.75 to 2.5 a from the center line of the nearest main bearing.

As far as the crank pins and the crank webs are concerned, it is usually sufficient to check them for a maximum load equal to the explosion load P_z (crank angle $\alpha=0$). A simple comparison of the expansion line of Fig. 33 with the tangential effort line of Fig. 178, will show that P_z decreases at a much faster rate than T increases, and since the moment arm $\frac{a}{2}$ for P_z is always greater than the arm r for T, it is plain that the shaft parts mentioned must be under greatest stress at the moment of explosion. This applies even to constant-pressure engines (of which the Diesel is a type) with cut-off as late as 12–15% of the stroke. Only the diameter of the main journal transmitting the power is determined by the forces existing at the moment of maximum tangential effort ($\alpha \sim 40°$). Nevertheless it is well to check the crank webs of the larger machines also in this position of the crank, and for machines having a heavy fly-wheel this computation becomes necessary. In what follows this double check has been used, in order to give a better insight into all the conditions that may prevail.

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[1] TRANSLATOR'S NOTE. This view is apparently not shared by some American builders, who are using the Tangye type of engine frame with its overhung crank.

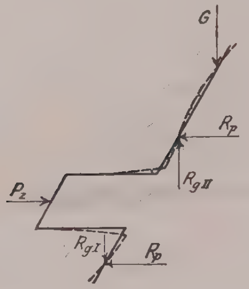


FIG. 180.

(a) **Maximum Piston Pressure P_z in the Dead Center Position.** In horizontal engines[1] in which part of the weight of the fly-wheel is not supported by an outboard bearing (the most unfavorable case) the **crank pin** is acted upon by the following moments:

A bending moment M_{b_1}, due to the piston pressure P_z.

A bending moment M_{b_2} due to the fly-wheel weight G and a torsion moment M_d, also due to G.

The moment M_{b_1} can at once be determined from the pressure P_z. The other two are most easily found by use of the reaction forces at the bearings. These reactions are as follows (for notation see Fig. 180):

Due to P_z, $$R_p = \frac{P_z}{2} \text{ lbs.} \qquad (26)$$

Due to G, $$R_{gI} = G\frac{a_1}{a}, \quad \text{and} \quad R_{gII} = R_{gI} + G = G\frac{a+a_1}{a} \text{ lbs.} \qquad (27)$$

Consequently $$G + R_{gI} - R_{gII} = 0.$$

$$M_{b_1} = \frac{P_z a}{4} \text{ in.-lbs.}; \qquad (28)$$

$$M_{b_2} = R_{gI}\frac{a}{2} \text{ in.-lbs.}; \qquad (29)$$

$$M_d = R_{gI} r \text{ in.-lbs.} \qquad (30)$$

Since M_{b_1} and M_{b_2} are acting at right angles, the combined bending moment will be

$$M_b = \sqrt{M^2_{b_1} + M^2_{b_2}} \text{ in.-lbs.}$$

The two moments M_b and M_d are next to be combined according to Bach's formula (see p. 168, Table 15). The resulting moment is

$$M_r = .35\, M_b + .65\sqrt{M^2_b + \alpha_0\, M^2_d} \text{ in.-lbs.}$$

---

[1] For modifications in the method of crank-shaft computations for vertical engines, see p. 176.

The factor $\alpha_0 = \frac{K_b}{1.3K_d}$. For good steel $K_b \sim 12000\text{–}13000$ lbs. per sq.in., and $K_d \sim 10000$ lbs. per sq.in., hence

$$\alpha_0 = \frac{13000}{1.3 \times 10000} \sim 1.0,$$

so that α_0 may hereafter be neglected.

From the above, the maximum bending stress in the crank pin will be

$$\sigma_b = \frac{M_r}{W} \text{ lbs. per sq.in.} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad (31)$$

Now, for solid crank pins,

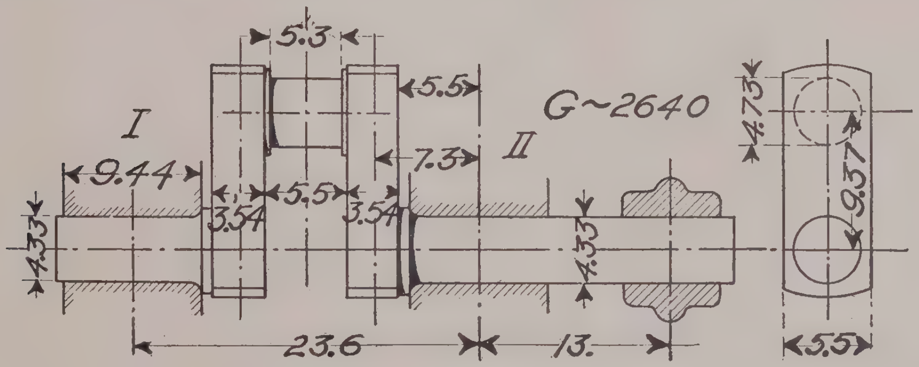
$$W = \frac{\pi d^3}{32} \sim \frac{d^3}{10}.$$

Hence

$$\sigma_b = \frac{10\,M_r}{d^3} \text{ lbs. per sq.in.} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad (31a)$$

The following example is inserted to make clear the relation existing between M_{b_1}, M_{b_2}, and M_d.

Example. 1. The crank shaft shown in Figs. 181 and 182 belongs to a Körting engine having a cylinder diameter of 10″. The number of revolutions is 160. With an explosion pressure of $p_z \sim 350$ lbs. per sq.in., the explosion load is $P_z = 27000$ lbs. The weight of fly-wheel $= G = 2640$ lbs. Then, from the above equations,



FIGS. 181 and 182.—Dimensions in Inches.

$$R_p = .5 \times 27000 = 13500 \text{ lbs.};$$

$$R_{g\text{I}} = 2640\frac{13}{23.6} = 1450 \text{ lbs.};$$

$$R_{g\text{II}} = R_{g\text{I}} + G = 1450 \overset{+}{*} 2640 = 4090 \text{ lbs.};$$

Hence

$$\left.\begin{aligned} M_{b_1} &= \frac{27000 \times 23.6}{4} = 160000 \text{ in.-lbs.} \\ M_{b_2} &= 1450 \times \frac{23.6}{2} = 17100 \text{ in.-lbs.} \end{aligned}\right\} M_b = \sqrt{160000^2 + 17100^2} = 161000 \text{ in.-lbs.}$$

$$M_d = 1450 \times 9.37 = 13\,600 \text{ in.-lbs.};$$

$$M_r = .35 \times 161\,000 + .65\sqrt{161\,000^2 + 13\,600^2} = 161\,500 \text{ in.-lbs.}$$

The *maximum bending stress* in the crank pin in the dead center position is therefore

$$\sigma_b = \frac{161\,500 \times 10}{4.73^3} = 15\,140 \text{ lbs. per sq.in.}$$

Neglecting the two moments M_{b_2} and M_d, the bending stress due to M_{b_1} alone would have been,

$$\sigma_b = \frac{160\,000 \times 10}{4.73^3} = 14\,960 \text{ lbs. per sq.in.}$$

This example shows that it is usually sufficient to check the crank-pin against the load P_z only, provided the weight G and the arm a are not abnormally great. If, however, M_{b_2} and M_d are neglected, we may derive the following simple expression:

$$\sigma_b = \frac{P_z a}{W4} = \frac{P_z a 10}{d^3 4} = \frac{2.5\,P_z a}{d^3} \text{ lbs. per sq.in.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (32)$$

Assuming $\sigma_b \leqq K_b = 14\,000$ lbs. per sq.in. for good open-hearth steel, the diameter of the crank-pin will then be

$$d \geqq \sqrt[3]{\frac{P_z a 2.5}{14\,000}} = \sqrt[3]{\frac{P_z a}{5600}} \text{ in.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (33)$$

Assuming further that $a \sim 1.8\,D$, which is approximately true for the larger engines, and putting $p_z = 350$ lbs. per sq.in. for our standard diagram, we will finally have the special equation

$$\mathbf{d} = \sqrt[3]{\frac{.785 \times 350 \times D^2 \times 1.8\,D}{5600}} = \sqrt[3]{.09\,D^3} = .448\,D \sim \mathbf{.45\,D}. \quad . \quad . \quad . \quad . \quad (33n)$$

The limit of safe stress above assumed, i.e., $\sigma_b \leqq 14\,000$ lbs. per sq.in., is apparently rather high, but in view of the fact that only the best kind of open-hearth steel should be used for crank-shafts, this stress may be considered safe, provided no other disturbing deformations occur. In the real case, also, the stresses are not quite as severe as the above safe method of computation would indicate. For instance, the real points of support of the shaft, as far as the explosion load on the pin is concerned, are not located in the middle planes of the bearings, but somewhere near the inner edge. For this reason the real free length concerned in the determination of the bending moment is not the distance a, but approximately $2c$ (see Fig. 179). Generally, $2c = .65$–$.75a$, hence the real bending moment is from 25–33% smaller than the computation with a indicates. It should further be considered that the load P_z acts in a normal direction but a very short time, so that in a sense but little time for serious deformation of the metal is available. Lastly, the load P_z in reality is not a concentrated load, as above assumed, but is distributed over the length of pin $= l$. With these modifications the bending moment due to P_z would then decrease from

$$M_{b_1} = \frac{P_z a}{4} \quad \text{to} \quad M_{b_1} = \frac{P_z}{2}\left(c - \frac{l}{4}\right). \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (28a)$$

In a similar manner the bending moments derived below for the crank webs and the main journals are in excess of the real stress conditions. All the computations are thus based upon the assumption of the most unfavorable conditions that can exist, hence the results give shafts which are safe not only against imperfections in manufacture and other accidents, but which combine great stiffness with sufficient strength.

The **Crank Webs or Arms,** at the moment of explosion, besides being under a purely compressive stress of minor importance, are also subject to bending and torsional stresses. The crank web nearest the fly-wheel or belt pulley is naturally under the greater stress. In this web we have the following moments (see Figs. 179 and 180):

A bending moment, due to P_z, $=M_{b_1}=\frac{P_z}{2}e$ in.-lbs. . . . . (34)

A bending moment, due to R_{g1}, $=M_{b_2}=R_{g1}\left(r-\frac{d}{2}\right)$ in.-lbs. . . . . (35)

and a torsion moment, due to R_{g1}, $=M_d=R_{g1}(a-e)$ in.-lbs. . . . . (36)

The stresses due to these moments are:

In bending, $\sigma_{b_1}=\frac{M_{b_1}}{\frac{1}{6}b^2h}$ lbs. per sq.in. . (37) and $\sigma_{b_2}=\frac{M_{b_2}}{\frac{1}{6}bh^2}$ lbs. per sq.in. . (38)

In torsion, $\tau=\frac{M_d}{\frac{2}{9}b^2h}$ lbs. per sq.in. . . . . (39)

Combining these stresses in the proper manner, we have:

Bending stress in the two edges under greatest stress due to $\sigma_{b_1}+\sigma_{b_2}=\sigma_b$;
Stress at the center line of the wide side of the arm due to σ_{b_1} and τ;
Stress at the center line of the narrow side of the arm due to σ_{b_2} and $\tau_1=\tau\frac{b}{h}$.

The bending and torsion stresses in the last two cases are combined into a single resultant stress, according to Bach's equation,

$\sigma_r=.35\sigma_b+.65\sqrt{\sigma_b^2+4\tau^2}$ lbs. per sq.in. . . . . (40)

This equation may be re-written

$$\frac{\sigma_r}{\sigma_b}=.35+.65\sqrt{1+4\left(\frac{\tau}{\sigma_b}\right)^2},$$

and to simplify the computation of σ_r the following table gives values for $\frac{\sigma_r}{\sigma_b}$ when the value of $\frac{\tau}{\sigma_b}$ is known.

TABLE 15

$\frac{\tau}{\sigma_b}=$	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9	1.0	1.1
$\frac{\sigma_r}{\sigma_b}=$	1.013	1.050	1.108	1.182	1.269	1.364	1.468	1.576	1.688	1.803	1.921
Ppt=	*3.7*	*5.8*	*6.4*	*8.7*	*9.5*	*10.4*	*10.8*	*11.2*	*11.5*	*11.8*	*11.9*

$\frac{\tau}{\sigma_b}=$	1.2	1.3	1.4	1.5	1.6	1.7	1.8	1.9	2.0	2.5	3.0
$\frac{\sigma_r}{\sigma_b}=$	2.040	2.160	2.282	2.405	2.529	2.653	2.778	2.904	3.030	3.664	4.300
Ppt=	*12*	*12.2*	*12.3*	*12.4*	*12.4*	*12.5*	*12.6*	*12.6*	*12.7*	*12.7*	*12.8*

The figures in italics in the lines marked *Ppt* may be used for interpolation. For example, suppose $\frac{\tau}{\sigma_b}=.52$, then the corresponding value of $\frac{\sigma_r}{\sigma_b}$ interpolated between the values given in the fifth and sixth columns of the above table may be found as follows:

$Ppt=2\times 9.5=19$, since .52 exceeds .50 by 2 in the last place. Then

$$\frac{\sigma_r}{\sigma_b}=\left\{\begin{matrix}1.269\\ .019\end{matrix}\right\}=1.288.$$

Example. 2. The right crank web of the crank-shaft, shown in Figs. 181 and 182, is under stress due to the following moments:

$$M_{b_1}=\frac{27{,}000}{2}\times 7.30=98600 \text{ in.-lbs.};\qquad \sigma_{b_1}=\frac{6\times 98600}{3.54^2\times 5.5}=8600 \text{ lbs. per sq.in.};$$

$$M_{b_2}=1450(9.37-2.37)=10150 \text{ in.-lbs.};\qquad \sigma_{b_2}=\frac{6\times 10150}{3.54\times 5.5^2}=570 \text{ lbs. per sq.in.};$$

$$M_d=1450(23.6-7.30)=23600 \text{ in.-lbs.};\qquad \tau=\frac{9\times 23600}{2\times 3.54^2\times 5.5}=1540 \text{ lbs. per sq.in.}$$

From the above, the stresses will combine as follows:

In the two edges of the arm under greatest stress, 8600 and 570 lbs. per sq.in.;

In the wide side of the arm, 8600 lbs. per sq.in. tension and 1540 lbs. per sq.in. shear.;

In the narrow side of the arm, 570 lbs. per sq.in. tension and $1540\times\frac{3.54}{5.5}=925$ lbs. per sq.in. shear.

The resultant maximum stresses therefore are:

Edges of arm: $\sigma_{b_1}+\sigma_{b_2}=8600+570=9170$ lbs. per sq.in.;

Wide side h: $\sigma_r=.35\times 8600+.65\sqrt{8600^2+[4\times 1540^2]}=8960$ lbs. per sq.in.;

Narrow side b: $\sigma_r=.35\times 570+.65\sqrt{570^2+[4\times 925^2]}=1450$ lbs. per sq.in.

It is clear at once that the stress in the narrow side of the arm is considerably below that occurring in the wide side. The thickness of the arm (b), in the direction parallel to the shaft axis, should be made quite liberal when the fly-wheel is seated on the shaft outside of the main bearings and no outboard bearing is used, because the explosion load may cause deflections in the crank arm which at once manifest themselves by unpleasant lateral vibrations in the fly-wheel rim. According to a well-tried rule of thumb, the crank arm thickness in center crank-shafts should be at least

$$b = .6 \text{ to } .7\, d.$$

The fate of a center crank-shaft too weak in the crank web is well shown by Fig. 183. This shaft was part of an engine of 20 H.P., with crank shaft above the cylinder, which made it a comparatively large engine of its type. The lateral vibrations of the fly-wheel rim were at first but slight, but during ten years of operation they increased in magnitude to such an extent that the engine could no longer be used.

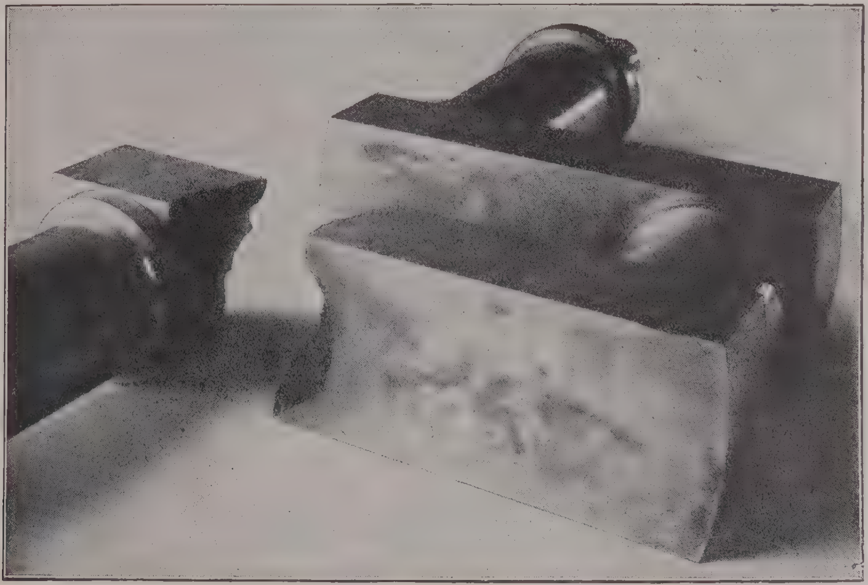


Fig. 183.—Broken Shaft from 20 H.P. Vertical Engine.

An investigation showed that the crank arm transmitting the power was ruptured through about four fifths of its cross-section near the junction with the main bearing, and that only the inner one fifth of the cross-sectional area remained sound. A part of the area of rupture had already lost its usual granular appearance and appeared polished, showing that friction between the surfaces must have occurred during operation. As will be seen from the figure, the fracture starts exactly in the joint between the journal and the arm, from which we can conclude without doubt that the sudden transition from one surface to the other was at least partly to blame for the occurrence. It points out the necessity of a liberal fillet at that point, as in the case of all other similar constructions under stress.

If the weight of the fly-wheel is supported by an outboard bearing, the moments M_{b_2} and M_d, due to G, disappear. The stresses in the crank arm are then only those due to P_z, that is, a certain compression stress and the stress due to moment M_{b_1}. For the latter, eq. (34) may be rewritten for our standard indicator diagram

$$M_{b_1}=140\ D^2e \text{ in.-lbs.}, \qquad (34n)$$

from which

$$\sigma_b=\frac{840\ D^2e}{b^2h}. \qquad (41)$$

The dimensions of the **main journal** transmitting the load are always determined by the maximum turning moment.

(*b*) **Maximum Turning Moment for** $\alpha=40°$. The piston pressure, P_t, acting in this position of the crank (see Fig. 184), causes a pressure T at right angles to the crank, and a pressure Z in the plane of the crank. From these the stresses acting in the various parts of the shaft may be derived, neglecting the moment due to G (see above). According to eq. (13*n*) and (14*n*), p. 159, $P_t=.7\ P_z$, and $T=.5\ P_z$. Hence the radial force Z with $\alpha=40°$ and $\beta=7.3°$, equals

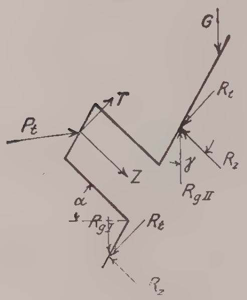


FIG. 184

$$Z=P_t \cos (40+7.3) \sim .68\ P_t \text{ lbs.},$$

or

$$Z=.68 \times .7\ P_z= \sim .48\ P_z \text{ lbs.} \qquad (42)$$

Of this amount one half $=.24\ P_z$, falls to each bearing.

The exact computation would have to be made, taking into account the reactions, R_t, R_z, and R_g, resulting from T, Z, and G, but since Example 1 showed that the stresses in the crank pin are not seriously increased by the fly-wheel weight G, this computation will for the sake of clearness not be made here.

The crank-pin is affected by the following moments, see Figs. 179 and 184.

$$\text{A bending moment due to } T,\quad M_{b_1}=\frac{Ta}{4}=\frac{R_ta}{2}=.125\ P_za. \qquad (43)$$

$$\text{A bending moment due to } Z,\quad M_{b_2}=\frac{Za}{4}=\frac{R_za}{2}=.120\ P_za. \qquad (44)$$

$$\text{A twisting moment due to } R_t,\quad M_d=R_tr=.25\ P_zr. \qquad (45)$$

M_{b1} and M_{b2} act at right angles, hence

$$M_b=\sqrt{M_{b_1}^2+M_{b_2}^2}=\sqrt{(.125\ P_za)^2+(.120\ P_za)^2}. \qquad (46)$$

From the moments as above computed we will then have the following stresses

$$\text{Bending,}\quad \sigma_b=\frac{10\ M_b}{d^3} \text{ lbs. per sq.in.} \qquad (47)$$

$$\text{Torsion,}\quad \tau=\frac{5\ M_d}{d^3}=\frac{1.25\ P_zr}{d^3} \text{ lbs. per sq.in.} \qquad (48)$$

The resultant stress will be, as before,

$$\sigma_r=.35\sigma_b+.65\sqrt{\sigma_b^2+4\tau^2}\leq K_b.$$

Example. 3. For the crank shaft of the Körting engine, treated in Example 1, $P_t = .7 \times 27\,000 = 19\,000$ lbs., $T = .5 \times 27\,000 = 13\,500$ lbs., and $Z = .48 \times 27\,000 = 13\,000$ lbs. Then

$$M_{b_1} = \frac{13\,500 \times 23.6}{4} = .125 \times 27\,000 \times 23.6 = 80\,000 \text{ in.-lbs.}$$

$$M_{b_2} = \frac{13\,000 \times 23.6}{4} = .120 \times 27\,000 \times 23.6 = 76\,500 \text{ in.-lbs.};$$

from which $$M_b = \sqrt{80\,000^2 + 76\,500^2} = 111\,600 \text{ in.-lbs.};$$

also $$M_d = .25 \times 27\,000 \times 9.37 = 63\,300 \text{ in.-lbs.}$$

Moment resulting from combination of M_b and M_d is

$$M_r = .35 \times 111\,600 + .65\sqrt{111\,600^2 + 63\,300^2} = 123\,000 \text{ in.-lbs.}$$

The maximum bending stress is therefore

$$\sigma_r = \frac{10 \times 123\,000}{4.73^3} = 11\,600 \text{ lbs. per sq.in.}$$

The computation carried through for the crank-pin in the dead center position, in Example 1, gave $\sigma_b = 14\,960$ lbs. per sq.in., neglecting G, which value is about one third larger than that found above.

For the crank-pin computation, as above carried out, the resolution of P_t into T and Z is superfluous, since we can write directly

$$M_b = \frac{P_t a}{4} = \frac{.7\, P_z a}{4} = .175\, P_z a. \quad . \quad . \quad . \quad . \quad . \quad . \quad (46a)$$

The same result may be derived by simplifying the radical of eq. (46). But since the values of T and Z must be known to compute the crank webs, it is better to resolve P_t at the outset.

The crank arm next to the fly wheel, if called upon, as usual, to transmit the entire power, is under compression due to the load $\frac{Z}{2}$, hence the stress is

$$\sigma = \frac{.5\, Z}{bh} \text{ lbs. per sq.in.} \quad . \quad . \quad . \quad . \quad . \quad . \quad (49)$$

This may, however, in most cases be neglected. The arm is further under a bending stress due to R_z, for which

$$M_{b_1} = R_z e = .24\, P_z e \text{ in.-lbs.,} \quad . \quad . \quad . \quad . \quad . \quad . \quad (50)$$

and $$\sigma_{b_1} = \frac{M_{b_1}}{\frac{1}{6} hb^2} = \frac{1.44\, P_z e}{hb^2} \text{ lbs. per sq.in.} \quad . \quad . \quad . \quad . \quad . \quad . \quad (51)$$

There is also a bending stress due to T, for which

$$M_{b_2} = T\left(r - \frac{d_{\mathrm{II}}}{2}\right) = .5\ P_z\left(r - \frac{d_{\mathrm{II}}}{2}\right) \text{ in.-lbs.}, \quad \dots \quad (52)$$

and
$$\sigma_{b_2} = \frac{M_{b_2}}{\frac{1}{6}bh^2} = \frac{3\ P_z\left(r - \frac{d_{\mathrm{II}}}{2}\right)}{bh^2} \text{ lbs. per sq.in.} \quad \dots \quad (53)$$

Lastly, there is a shearing stress due to R_t, for which

$$M_d = R_t e = .25\ P_z\, e, \quad \dots \quad (54)$$

and
$$\tau = \frac{M_d}{\frac{2}{9}b^2h} = \frac{1.13\, P_z e}{b^2h} \text{ lbs. per sq.in.} \quad \dots \quad (55)$$

These various stresses combine, as on p. 168, as follows:
In two edges of the arm, σ_{b_1} and σ_{b_2}.
In the center of the wide side of the arm, σ_{b_1} and τ.
In the center of the narrow side of the arm σ_{b_2} and $\tau_1 = \tau\frac{b}{h}$.

The bending stresses σ_b, and the shearing stresses τ, should next be combined according to eq. (40), or Table 15, p. 168, into one resultant stress which must remain smaller than the allowable bending stress K_b for the material. The elastic deformations of the shaft due to the weight G or the reaction $R_{g\mathrm{II}}$ tend to relieve the arm next to the wheel (see Fig. 180, p. 164), but the determination of the amount of stress reduction due to this cause can not be made with any certainty.

If the power of the engine is taken off equally on both sides of the crank, a condition which can not be maintained with any certainty, each crank arm is subject to the stress caused by .5 T, so that M_{b_2} and σ_{b_2} will be only one half of what they were before.

Example. 4. Considering again the crank shaft treated in the previous examples, the stress conditions are the following:

$$\sigma_{b_1} = \frac{1.44 \times 27\,000 \times 7.3}{5.5 \times 3.54^2} = 4100 \text{ lbs. per sq.in.};$$

$$\sigma_{b_2} = \frac{3 \times 27\,000(9.37 - 2.17)}{3.54 \times 5.5^2} = 5450 \text{ lbs. per sq.in.};$$

$$\tau = \frac{1.13 \times 27\,000 \times 7.3}{3.54^2 \times 5.5} = 3240 \text{ lbs. per sq.in.}$$

Combining these stresses we have:

In two edges of the arm 4100+5450=9550 lbs. per sq.in. in tension. At the center of the wide side h, 4100 lbs. per sq.in. in tension and 3240 lbs. per sq.in. in shear. At the center of the narrow side b, 5450 lbs. per sq.in. in tension and $3240 \times \frac{3.54}{5.50} = 2080$ lbs. per sq.in. in shear. The maximum stress in this case, as in the dead center position, is found in the edge of the arm. By combining σ_b and τ into a resultant stress σ_r, for each side of the arm, as per eq. (40), we find the maximum stress in the wide side, $h = 6250$ lbs. per sq.in., and in the narrow side $b = 6320$ lbs. per sq.in.

The dangerous section in the **main journal** transmitting the power is close to the outside face of the crank arm, that is, close to the junction of the crank arm with the shaft. If no outboard bearing is used, this section is under a bending stress due to Z and G, while T causes both bending and twisting. The shearing stresses due to Z are of minor importance and may be neglected.

In the case of horizontal machines[1] the reactions R_z and R_{gII} enclose an angle $\gamma = 90 - 40 = 50°$ (see Fig. 184). The resultant of these forces may therefore be computed from the formula

$$R_{\text{res.}} = \sqrt{R_z^2 + R_{gII}^2 \pm 2 \cos \gamma R_z R_{gII}},$$

in place of using the triangle of forces. When $\gamma = 40°$, $\cos 40° = .6428$, and $2 \cos \gamma = 1.285$, hence for this case

$$R_{\text{res.}} = \sqrt{R_z^2 + R_{gII}^2 - 1.285\ R_z R_{gII}}.$$

Where the computation must be very accurate, however, it should be noted that the bending moment due to the fly-wheel in this case is

$$= G(a_1 + e_1) - R_{gII} e_1.$$

The reaction resulting from this is therefore

$$R_g = \frac{G(a_1 + e_1) - R_{gII} e_1}{e_1} = G \frac{a_1 + e_1}{e_1} - R_{gII}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (56)$$

This, together with the reaction due to Z and T, put the journal under bending stress. If, instead of considering the reaction due to Z and T separately, we again introduce the connecting-rod pressure P_t, as was done on p. 171, the resultant force may be easily found from the equation,

$$R_{\text{res.}} = \sqrt{R_s^2 + R_g^2 + 2\ R_s R_g \sin 7.3°} = \sqrt{R_s^2 + R_g^2 + .254\ R_s R_g}, \quad . \quad . \quad . \quad . \qquad (57)$$

in which R_s stands for the reaction due to P_t, and is equal to $\frac{P_t}{2}$.

Hence the resultant bending moment is

$$M_{b\ \text{res.}} = R_{\text{res.}}\ e_1. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (57a)$$

The torsional moment due to T is

$$M_d = Tr = 0.5\ P_z r. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (58)$$

These two moments combine to give the maximum stress in the journal.

In eq. (57) 7.3° is the connecting-rod angle β, at the instant of maximum turning moment, for which $\alpha = 40°$. Sin 7.3° therefore corresponds to the cosine of the angle between R_s and R_g $(90 + 7.3°)$ in the force triangle.

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[1] For modifications regarding vertical machines, see p. 176.

From the combined moment

$$M_r = .35\ M_{b\,\text{res.}} + .65\sqrt{M_b{}^2_{\text{res.}} + M_d{}^2},$$

we derive, as before, the maximum bending stress

$$\sigma_r = \frac{M_r}{W} \text{ lbs. per sq.in.,}$$

from which, for solid shafts

$$\sigma_r = \frac{10\ M_r}{d_{\text{II}}^3} \text{ lbs. per sq.in.}$$

The above computation is based on the assumption that the journal is called upon to resist the entire twisting moment T_r. This is really the case, even when the power is all transmitted through one journal, only under the most unfavorable circumstances, as, for instance, when the engine is just starting up. In normal operation, the moment of inertia of the fly-wheel

$$I = \Sigma M r_s^2 = \frac{GD_s^2}{4g} \sim \frac{GD_s^2}{128.8}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (59)$$

tends to relieve the stress in the dangerous section of the journal, so that the twisting moment existing at any time is the difference between T_r and the moment due to the wheel. But since the shaft must meet the highest maximum stresses that can occur, the computation should be made with the entire twisting moment T_r.

The middle cross-section of the journal, assuming the shaft perfectly rigid, is only under stress due to the twisting moment T_r, while the bending stresses due to G and Z cancel each other. In any actual case, of course, perfect rigidity does not exist, which means that there will also be bending stresses in that cross-section of the journal. Owing to the uncertainty existing with reference to the magnitude of the moments involved, an exact determination of these bending stresses is not possible. The shaft, however, is safe if the journal is given a diameter throughout its entire length equal to that computed above for the diameter near the junction of shaft and crank web.

Example. 5. Take again the crank-shaft of the previous examples. Figs. 181 and 182. The reactions.

$$R_s = \frac{P_t}{2} = \frac{19\,000}{2} = 9500 \text{ lbs.,} \quad \text{and} \quad R_{g\text{II}} = 4090 \text{ lbs.}$$

Since $G = 2640$ lbs., we have from eq. (56),

$$R_g = \frac{2640(13.0 + 5.5)}{5.5} - 4090 = 4800 \text{ lbs.}$$

Hence from eq. (57) and those following

$$R_{\text{res.}} = \sqrt{9500^2 + 4800^2 + .254 \times 9500 \times 4800} = 11\,200 \text{ lbs.;}$$

$$M_{b\,\text{res.}} = 11\,200 \times 5.5 = 61\,600 \text{ in.-lbs.;}$$

$$M_d = \frac{27\,000}{2} \times 9.37 = 126\,500 \text{ in.-lbs.}$$

Combining $M_{b\text{ res.}}$ and M_d, we have

$$M_r=.35\times61600+.65\sqrt{61600^2+126500^2}=113000 \text{ in.-lbs}$$

From this the combined fiber stress then is

$$\sigma_r=\frac{10\,M_r}{d_{\text{II}}^3}=\frac{10\times113000}{4.33^3}=13800 \text{ lbs. per sq.in.}$$

In case an *outboard bearing* is employed, the effect of the fly-wheel weight G disappears as far as the section at the junction between shaft and crank arm is concerned, and d_{II} depends solely upon the piston pressure P_z or P_t. The length of shaft a_1+a_2, between the main and outboard bearings is, however, under stress due to the weight of the fly-wheel mainly, but also has to take care of whatever stress there may be due to the belt pull. As far as this part of the shaft is concerned, the influence of the turning effort on the stresses is of secondary importance as compared with the bending stresses due to G and the belt pull, even if the equalizing action that the rotating masses possess, owing to their inertia, be left out of account.

The maximum deflection occurring at the center of the wheel seat may be determined from

$$f=\left(G+\frac{l^2+ab}{8ab}G_w\right)\frac{a^2b^2}{3EIl} \text{ in.}, \quad \ldots \ldots \ldots \quad (60)$$

in which $l=a+b=$distance between the middle cross-sections of the second main bearing and the outboard bearing [referring to Fig. 179, p. 163, $l=a_1+a_2$, $a=a_1$, and $b=a_2$, G_w is the weight in pounds of a length of shaft equal to l, $E=$the modulus of elasticity ($=30000000$ for O–H steel), and $I=$moment of inertia of the shaft cross-section]. The belt pull may be directly determined from the tangential force required to transmit the necessary horse-power at the given peripheral speed. That is,

$$P=\frac{550\,N_{\max}}{v}=\frac{60\times550\,N_{\max}}{\pi nD_s}\sim\frac{11000\,N_{\max}}{nD_s} \text{ lbs.}, \quad \ldots \ldots \quad (61)$$

in which $N_{\max}$ is the actual maximum brake horse-power, v the velocity of belt or rope in feet per second, $n=$r.p.m., and $D_s=$the diameter of pulley or sheave in feet. In order to possess sufficient friction to transmit the load, the belt must have a certain initial tension

$$S=\frac{P}{x-1} \text{ lbs.}, \quad \ldots \ldots \ldots \ldots \quad (62)$$

so that the total pull at right angles to the shaft will be

$$A=P\frac{x+1}{x-1} \text{ lbs.} \quad \ldots \ldots \ldots \ldots \quad (63)$$

The factor x depends upon the kind of belt or rope and the surface of the pulley or sheave, also upon the arc of contact, i.e., upon the ratio of transmission. For leather belts and nearly equal diameter of driver and driven pulley, we may take the load on the shaft due to the total pull equal to

$$A=6.5 \text{ to } 9\,P \text{ lbs.}$$

Since the bending moment due to G is vertical, while that due to A is in the majority of cases nearly horizontal, both are to be combined into a single resultant moment. The maximum total stress in the shaft length a_1+a_2 finally is that due to this resultant moment combined with that due to the turning moment T_r.

It should be noted that the maximum stress in the section under discussion should be kept low, in order to prevent any noticeable deflections, which are sure to cause hot bearings and lateral vibrations of the fly-wheel rim. Further, it should not be forgotten that the shaft at this section is usually weakened by key-seats.

In *vertical engines* with crank shafts $\left\{\begin{matrix}\text{below}\\ \text{above}\end{matrix}\right\}$ the cylinder, P_z and G have $\left\{\begin{matrix}\text{the same direction}\\ \text{opposite directions}\end{matrix}\right\}$, which affects friction and strength conditions $\left\{\begin{matrix}\text{unfavorably}\\ \text{favorably}\end{matrix}\right\}$. If no outboard bearing is employed, the bending moment, resulting from the combination of P_z and G, in the main journal transmitting the power, is

$$M_b=M_{b_1}\pm M_{b_2}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (64)$$

in which the plus sign applies to vertical engines with crank shaft below the cylinder, and the minus sign to engines with shaft above the cylinder. The influence of G on the crank pin and the crank arms is opposite to that above indicated, assuming that sensible deflections do not occur (see Example 7 below).

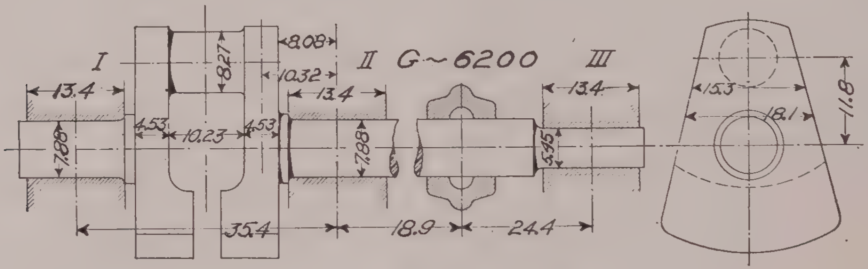


FIG. 185.—Dimensions in Inches.

At crank position for maximum turning effort ($\alpha=40°$ and $\beta=7.3°$), the main journal transmitting the power will be under stress due to a force resulting from the combination of rod pressure P_t, and fly-wheel weight G, equal to

$$R_s=\sqrt{R_s^2+R_g^2\mp 2\,R_sR_g\cos 7.3°}=\sqrt{R_s^2+R_g^2\mp 1.984\,R_sR_g}. \quad . \quad . \quad . \quad . \quad (65)$$

In this equation, when the shaft is above the cylinder, the plus sign applies, while for an engine with the shaft below the cylinder (the ordinary type of vertical engine) the minus sign is to be used. This is contrary to eq. (64).

The determination of the twisting moment M_d and all other strength computations are made the same as for horizontal machines.

Example. 6. Fig. 185 shows the crank shaft of a 50 H.P. vertical Bánki engine. Cylinder diameter $D=15.75''$, stroke $S=23.6''$, $n=135$ r.p.m., $p_z\sim570$ lbs. per sq.in., and $G=6200$ lbs. On account of the unusually high explosion pressure and the rapid drop of the expansion line of the indicator diagram, it will be safer not to determine the reaction forces, etc., from the general equations given, but to determine them accurately from the proper diagrams.

As will be seen from Plate IV, Fig. 16, the maximum turning moment occurs much earlier in the stroke than is common ($\alpha=21°$, $\beta=4.1°$); the pressure on the piston at this instant is $\sim .8\,P_z$, from which we can determine the following forces:

$$P_z=570\times .785\times 15.75^2=110000 \text{ lbs.};$$

$$P_t=.8\times 110000=88000 \text{ lbs.};$$

$$T=88000\,\frac{\sin\,(21+4.1)}{\cos\,4.1}\sim 37400 \text{ lbs.},$$

$$Z=88000\,\frac{\cos\,(21+4.1)}{\cos\,4.1}=80000 \text{ lbs.}$$

Mean piston pressure of a complete cycle, also accurately determined

$$p_m=40 \text{ lbs. per sq.in., so that } P_m=40\times .785\times 15.75^2=7800 \text{ lbs.}$$

Bearing pressure due to G

$$\text{in II,}\quad R_{g\mathrm{II}}=6200\,\frac{24.4}{24.4+18.9}=3480 \text{ lbs.};$$

$$\text{in III,}\quad R_{g\mathrm{III}}=6200\,\frac{18.9}{24.4+18.9}=2680 \text{ lbs.}$$

(*a*) *Friction Losses:*

$$\text{Maximum pressure without reference to weight of shaft or to belt pull,}\quad \left\{\begin{array}{l}
K_{\max}=\dfrac{110000}{8.27\times 10.23}=1300 \text{ lbs. per sq.in., Crank Pin;}\\[2ex]
K_{\max}=\dfrac{.5\times 110000}{7.88\times 13.4}=505 \text{ lbs. per sq.in., Main Journal I;}\\[2ex]
K_{\max}=\dfrac{(.5\times 110000)+3480}{7.88\times 13.4}=555 \text{ lbs. per sq.in., Main Journal II;}\\[2ex]
K_{\max}=\dfrac{2680}{5.45\times 13.4}=37 \text{ lbs. per sq.in., Outboard Journal III.}
\end{array}\right.$$

$$\text{Mean pressure for complete cycle,}\quad \left\{\begin{array}{l}
K_m=\dfrac{7800}{8.27\times 10.23}=92.0 \text{ lbs. per sq.in., Crank Pin;}\\[2ex]
K_m=\dfrac{.5\times 7800}{7.88\times 13.4}=37.0 \text{ lbs. per sq.in., Main Journal I;}\\[2ex]
K_m=\dfrac{(.5\times 7800)+3480}{7.88\times 13.4}=69.5 \text{ lbs. per sq.in., Main Journal II;}\\[2ex]
K_m=\dfrac{2680}{5.45\times 13.4}=37 \text{ lbs. per sq.in., Outboard Journal III.}
\end{array}\right.$$

Circumferential speed v and work of friction $K_m v$:

Crank Pin, $v=\dfrac{8.27\times\pi\times135}{12\times60}=4.88$ ft. per sec.; $K_m v=92\times4.88=450$ ft.-lbs. per sec.

Main Journal I, $v=\dfrac{7.88\times\pi\times135}{12\times60}=4.67$ ft. per sec.; $K_m v=37\times4.67=173$ ft.-lbs. per sec.

Main Journal II, $v=\dfrac{7.88\times\pi\times135}{12\times60}=4.67$ ft. per sec.; $K_m v=69.5\times4.67=326$ ft.-lbs. per sec.

Outboard Bearing III, $v=\dfrac{5.45\times\pi\times135}{12\times60}=3.17$ ft. per sec.; $K_m v=37\times3.17=120$ ft.-lbs. per sec.

(*b*) *Strength Computations. Dead Center Position:*

$$\text{Crank pin: } M_b=\frac{110000\times35.4}{4}=968000 \text{ in.-lbs.,}\quad \sigma_b=\frac{968000}{.1\times8.27^3}=17200 \text{ lbs. per sq.in.}$$

On account of the very high explosion load, it is well to check the pin for shearing stress. This is in this case equal to

$$\tau=\frac{.5\times110000\times4}{53.8\times3}=1360 \text{ lbs. per sq.in.}$$

The combined stress therefore is

$$\sigma_r=.35\times17200+.65\sqrt{17200^2+4\times1360^2}=\mathbf{17350} \text{ lbs. per sq.in.}$$

Since without τ, σ_b was 17200 lbs. per sq.in., it is evident from the above that even in this case the shearing stress in the pin may be neglected.

Crank arms: Compressive stress $\sigma=\dfrac{.5\times110000}{15.3\times4.53}=800$ lbs. per sq.in.;

$$M_b=.5\times110000\times10.32=568000 \text{ in.-lbs.,}\quad \sigma_b=\frac{568000\times6}{4.53^2\times15.3}=10800 \text{ lbs. per sq.in.}$$

$$\sigma+\sigma_b=800+10800=\mathbf{11600} \text{ lbs. per sq.in.}$$

Junction of Main Journals I and II with the Crank Arms:

$$M_b=.5\times110000\times8.08=445000 \text{ in.-lbs.,}\quad \sigma_b=\frac{445000\times10}{7.88^3}=\mathbf{9100} \text{ lbs. per sq.in.}$$

Shearing stress τ may be neglected, as above.
Position of Maximum Turning Moment ($\alpha=21°$).

Crank Pin: The simplest way is to determine the bending stress from P_t (instead of T and Z). Then we shall have

$$M_b=\frac{88000\times35.4}{4}=780000 \text{ in.-lbs.,}\quad \sigma_b=\frac{780000\times10}{8.27^3}=13800 \text{ lbs. per sq.in.}$$

From the turning effort T we have

$$M_d=\frac{37\,400}{2}\times 11.8=221\,000 \text{ in.-lbs.,} \qquad \tau=\frac{221\,000\times 5}{8.27^3}=1940 \text{ lbs. per sq.in.}$$

Combining σ_b and τ, we have

$$\sigma_r=\mathbf{15\,740} \text{ lbs. per sq.in.,}$$

which is considerably smaller than the stress at the inner dead center position.

Crank arm transmitting the load:

From R_z: $$M_{b_1}=\frac{80\,000}{2}\times 10.32=414\,000 \text{ in.-lbs.,} \qquad \sigma_{b_1}=\frac{414\,000\times 6}{15.3\times 4.53^2}=7900 \text{ lbs. per sq.in.}$$

From T: $$M_{b_2}=37\,400\left(11.8-\frac{7.88}{2}\right)=295\,000 \text{ in.-lbs.,} \qquad \sigma_{b_2}=\frac{295\,000\times 6}{4.53\times 18.1^2}=1200 \text{ lbs. per sq.in.}$$

From R_t: $$M_d=\frac{37\,400}{2}\times 10.32=192\,000 \text{ in.-lbs.,} \qquad \tau=\frac{192\,000}{\frac{2}{9}\times 4.53^2\times 15.3}=2750 \text{ lbs. per sq.in.}$$

From $\frac{T}{2}$: Shear $\tau=\frac{3}{2}\times\frac{18\,700}{4.53\times 15.3}=405$ lbs. per sq.in.

From the above the combined stresses are:

Center of wide side of arm:

$$\sigma_{b_1}=7900 \text{ lbs. per sq.in.,} \qquad \tau_r=2750+405=3155 \text{ lbs. per sq.in.,}$$

from which $$\sigma_r=\mathbf{9300} \text{ lbs. per sq.in}$$

Center of narrow side of arm:

$$\sigma_{b_2}=1200 \text{ lbs. per sq.in.,} \qquad \tau_r=\sqrt{\left(2750\times\frac{4.53}{15.3}\right)^2+405^2}=910 \text{ lbs. per sq.in.,}$$

from which $$\sigma_r=\mathbf{1830} \text{ lbs.per sq.in.}$$

In the edge of the arm:

$$\sigma_b=7900+1200=9100 \text{ lbs. per sq.in.,} \qquad \tau=405 \text{ lbs. per sq.in.,}$$

from which $$\sigma_r=\mathbf{9130} \text{ lbs. per sq.in.}$$

Here again the stress is less than at the dead center position of the crank.

Main Journal transmitting the Load:

$$M_b=\frac{88\,000}{2}\times 8.08=355\,000 \text{ in.-lbs.,} \qquad \sigma_b=\frac{355\,000\times 10}{7.88^3}=7260 \text{ lbs. per sq.in.;}$$

$$M_d=37\,400\times 11.8=440\,000 \text{ in.-lbs.,} \qquad \tau=\frac{440\,000\times 5}{7.88^3}=4500 \text{ lbs. per sq.in.}$$

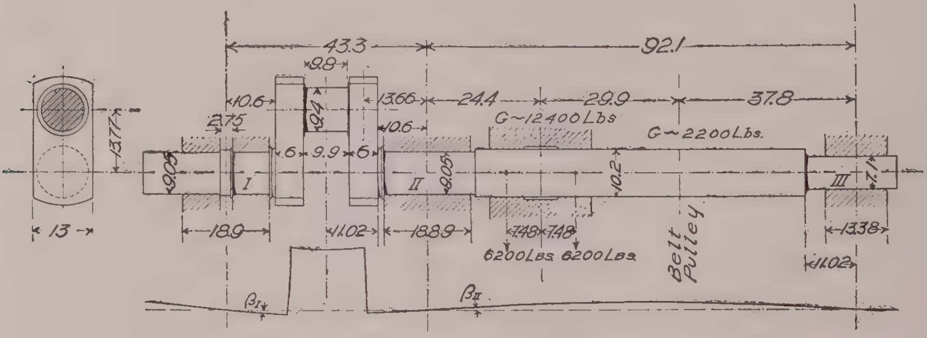
Combining these gives

$$\sigma_r=\mathbf{10100} \text{ lbs. per sq.in.,}$$

which is greater than the stress found for the same section in the dead center position. This example shows, as was stated at the outset, that the crank pin and the crank arms receive their maximum stress through the load P_z, while the maximum stress in the main journal transmitting the power is due to the moment T_r.

In vertical as well as in horizontal machines, the stresses occurring in the shaft are favorably or unfavorably affected by the elastic deformations of the shaft under load, depending upon whether the slope of the elastic curve is directed against or with the lines of action of the external forces. In general, the change is to make the real stresses less than those computed above on the assumptions set forth. This is especially true for those parts of the shaft where the maximum stresses may be expected to occur, that is, in the crank pin and the crank arm and main journal transmitting the load. The method used in the above computations therefore combines simplicity with great safety. The exact determination of the various stresses involved in the crank-shafts, taking into account the influence of deformation in the material, is quite complicated, and is therefore made only in very important cases as a check. The main advantage of such an investigation is that it gives more exact information as to the real pressures on the main bearings than it is possible to obtain with the simple but more usual method.

The approximate shape assumed by the elastic curve of a crank-shaft at the moment of explosion is shown, much exaggerated, in Fig. 187.



FIGS. 186 and 187.—Dimensions in Inches.

Example. 7. Figs. 186 and 187 show the crank-shaft of a 100 H.P. vertical Güldner engine. Maximum piston pressure is approximately 125000 lbs., weight of fly-wheel is about 12300 lbs., the belt pulley is about 220 lbs. From these weights we may derive the maximum stresses in the dangerous sections of the shaft as follows:

TABLE 16

	Dead Center Position.			Crank Position $\alpha=35°$.			
	Crank Pin.	Crank Arm Under Load.	Middle Bearing.	Crank Pin.	Crank Arm Under Load.	Middle Bearing.	Seat of Fly-wheel.
Principal stresses, lbs. per sq.in., according to							
(a) Simplified method	16 300	10 850	8100	10 500	6950	10 250	
(b) Strict method	14 400	8 800	4850	9 250	6100	8 400	5550

In the above table the values in line (*a*) were computed by use of the simplified method above developed, while the values of line (*b*) are based on principles laid down in the well-known thesis[1] of Ensslin.

The latter also furnishes the following values for the maximum deflection at the moment of maximum turning effort ($\alpha = 35°$): Slope $\beta_1 = \frac{1}{1120}$; $\beta_{II} = \frac{1}{4070}$; greatest deflection in the shaft next to the seat of the fly-wheel is less than .012 in.

The friction losses in this shaft are as follows:

TABLE 17

	Journal.			Crank Pin.
	I	II	III	
Maximum pressure, $K_{\max}$, lbs. per sq.in.	427	423	....	1340
Mean pressure, K_m, lbs. per sq.in.	47	98	48	146.5
Circumferential speed, v, ft. per sec.	6.28	6.32	4.95	6.57
Work of friction, $K_m v$, ft.-lbs. per sec.	296	620	238	960

The design of **multiple-throw crank-shafts** may be based on the methods of computation already developed. Since the maximum turning moment Tr, for each crank always occurs within the first one fifth of the expansion stroke, no two $T_{\max}$ can occur at the same time for any of the multicylinder combinations now used. For that reason the diameter of the shaft may in this case also be computed on the basis of one $T_{\max}r$, as in the previous instances. As for the rest of the dimensions, the greater free length of shaft (due to greater distances between bearings) cause greater bending stresses (due to P_z in the dead center position), which call for a corresponding increase in the dimensions. In the case of engines having four or more cylinders, the phases of the cycles in the various cylinders should be so arranged that, in any pair of cylinders having cranks 180° apart, expansion does not occur in one cylinder while compression takes place in the other. If this point is neglected it may happen that the turning moment due to the explosion in one cylinder is accompanied by the resisting moment of the compression in the other, which puts the intervening crank arm under stresses quite possibly excessive. The phases in the various cylinders of a four-cylinder machine, for instance, should be arranged as indicated in Fig. 188. The crank arrangement shown there is also the most favorable as regards balancing. The inclined position of the middle crank arm must be regarded as a make-shift only to which recourse must be had when the cylinders are so close together that there is no room for a bearing between them. In general this construction should be avoided, because the inclined arms are subject to the action of additional forces which seriously increase the stresses occurring, besides complicating the method of computation. The following example furnishes evidence regarding these points:

Example. Figs. 188 and 189 show the dimensions of the crank-shaft of a 4-cylinder 4-cycle marine engine whose cylinder diameter $= 9.85''$; stroke $= 12.55''$; and r.p.m. $= 350$. Taking the explosion pressure at 356 lbs. per sq.in., the load on each crank will be $P_z = 356 \times .785 \times 9.85^2$

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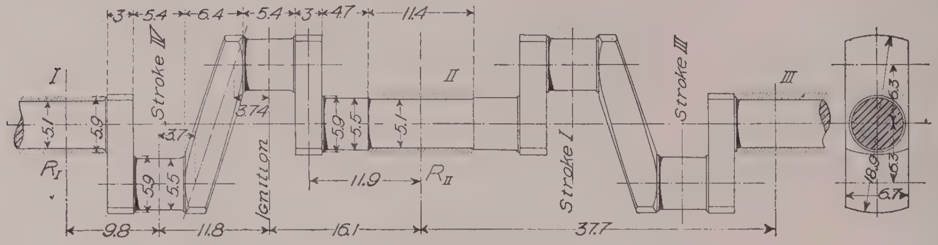
[1] Max Ensslin, Mehrmals gelagerte Kurbelwellen mit einfacher und doppelter Kröpfung, Stuttgart, 1902.

~ 27000 lbs., $P_t = .7 \times 27000 \sim 18000$ lbs., the maximum tangential force $T = .5 \times 27000 = 13500$ lbs., and the radial pressure $Z = .47 \times 27000 = 12700$ lbs. The weight of the light fly-wheel may be neglected. Assume that the second cylinder from the left is just exploding. Since the shaft is symmetrical on both sides of the middle line only one half needs to be checked.



FIGS. 188 and 189.—Dimensions in Inches

Crank Pin. *In dead center position:* Reactions due to P_z

$$\text{In bearing I,} \quad R_{\text{I}} = \frac{27000 \times 16.1}{37.7} = 11500 \text{ lbs.}$$

$$\text{In bearing II,} \quad R_{\text{II}} = \frac{27000 \times 21.6}{37.7} = 15500 \text{ lbs.}$$

Bending Moment on the Crank Pin:

$$M_b = 15500 \times 16.1 = 11500 \times 21.6 = 250000 \text{ in.-lbs.,} \qquad \sigma_b = \frac{250000}{.1 \times 5.5^3} = \mathbf{15\,100} \text{ lbs. per sq.in.}$$

Crank Position, $\alpha = 40°$. The connecting-rod force P_t causes a reaction in bearing II equal to

$$R_{\text{II}} = \frac{18000 \times 21.6}{37.8} = 10300 \text{ lbs.}$$

In this position, therefore, the crank pin is under the following stresses:

$$\text{Bending due to } R_{\text{II}}\text{:} \quad M_b = 10300 \times 16.1 = 166000 \text{ in.-lbs.} \qquad \sigma_b = \frac{166000}{.1 \times 5.5^3} = 9800 \text{ lbs. per sq.in.}$$

$$\text{Torsion due to } R_t\text{:} \quad M_d = 7750 \times 6.3 = 487000 \text{ in.-lbs.,} \qquad \tau = \frac{48700}{.2 \times 5.5^3} = 1440 \text{ lbs. per sq.in.}$$

Combining σ_b and τ, the resulting maximum stress is $\sigma_r = \mathbf{10\,100}$ lbs. per sq.in.

Crank Arm Transmitting Load. *Dead Center Position.* In this position the arm is under stress due to R_{I} or R_{II} only. Hence

$$M_b = 15500 \times 11.90 = 186000 \text{ in.-lbs.,} \qquad \sigma_b = \frac{186000 \times 6}{6.7 \times 3.0^2} = \mathbf{18\,600} \text{ lbs. per sq.in.}$$

Crank Position $\alpha=40°$: The reactions due to T and Z furnish the following stresses:

Bending due to R_z: $M_{b_1}=7280\times11.90=87000$ in.-lbs., $\sigma_{b_1}=\dfrac{87000\times6}{6.7\times3.0^2}=8700$ lbs. per sq.in.

Bending due to T: $M_{b_2}=13500\left(6.3-\dfrac{5.5}{2}\right)=48000$ in.-lbs., $\sigma_{b_2}=\dfrac{48000\times6}{3.0\times6.7^2}=2170$ lbs. per sq.in.

Torsion due to R_t: $M_d=7750\times11.90=93000$ in.-lbs., $\tau=\dfrac{93000}{\frac{2}{9}\times6.7\times3.0^2}=7000$ lbs. per sq.in.

Hence using Table 15, p. 168, we shall have the following combined stresses:

In the wide side of the arm

From $\sigma_{b_1}=8700$ lbs. per sq.in., and $\tau=7000$ lbs. per sq.in., $\sigma_r=\mathbf{13800}$ lbs. per sq.in.

In the narrow side of the arm

From $\sigma_{b_2}=2170$ lbs. per sq.in., and $\tau_1=7000\times\dfrac{3.0}{6.7}=3130$ lbs. per sq.in., $\sigma_r=\mathbf{4900}$ lbs. per sq.in.

In the edges of the arm

$$\sigma_b=\sigma_{b_1}+\sigma_{b_2}=8700+2170=\mathbf{10870}\text{ lbs. per sq.in.}$$

Again the maximum stress is considerably less in this position than in the dead center position.

The same method of computation would be applicable to the intermediate crank arm, if it had its normal position at right angles to the axis of the shaft. For the inclined arm, on the other hand, the method is as follows:

Intermediate Arm. *Dead Center Position.* The first step is to resolve the reactions at the bearings due to P_z into components parallel and at right angles to the center line of the inclined arm. This is most easily done graphically by means of the force polygon. Fig. 190 shows the method and needs no further explanation. It is sufficient also in this case to confine the computations to one of the arms. The dangerous sections to be considered are those at x–x and at y–y.

The moments acting on the former section are:

$$M'_{b_1}=14700\times20.8=306000\text{ in.-lbs.}$$

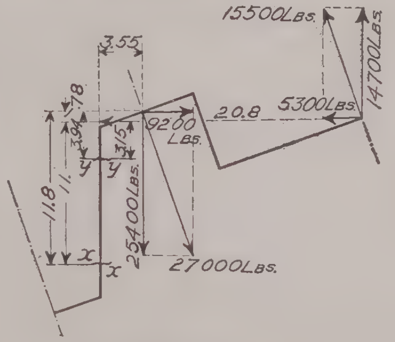


FIG. 190.—Dimensions in Inches.

Opposed to this moment, however, is that due to the component of P_z at the crank pin, which is

$$M''_{b_1}=25400\times3.55=90000\text{ in.-lbs.}$$

Hence the net moment is

$$M_{b_1}=306000-90000=216000\text{ in.-lbs.}$$

A second pair of moments acting on this section are

$$M'_{b_2}=9200\times11.8=109000\text{ in.-lbs.,}$$

and

$$M''_{b_2}=5300\times11=58000\text{ in.-lbs.}$$

These again are opposed to each other, and the net moment is

$$M_{b2} = 109\,000 - 58\,000 = 51\,000 \text{ in.-lbs.}$$

Now M_{b1} and M_{b2} also act in opposite directions, so that the net effective moment on x–x finally is

$$M_b = 216\,000 - 51\,000 = 165\,000 \text{ in.-lbs.}$$

The section modulus of the cross-section x–x, referred to the wide side of the arm as the base, is in this case $\frac{1}{6} \times 6.7 \times 3.82^2 = 16.1 \text{ in}^4$, so that the maximum stress in this section is

$$\sigma_b = \frac{165\,000}{16.1} = \mathbf{10\,200} \text{ lbs. per sq.in.}$$

Taking next the section y–y, we find the following:

$$\text{Moment } M'_{b1} = 14\,700 \times 20.8 = 306\,000 \text{ in.-lbs}$$

$$\text{Moment } M''_{b1} = 25\,400 \times 3.55 = 90\,000 \text{ in.-lbs.}$$

$$\text{Net moment } M_{b1} = 306\,000 - 90\,000 = 216\,000 \text{ in.-lbs.}$$

Further

$$\text{Moment } M'_{b2} = 9200 \times 3.94 = 33\,000 \text{ in.-lbs.}$$

$$\text{Moment } M''_{b2} = 5300 \times 3.15 = 16\,500 \text{ in.-lbs.}$$

$$\text{Net moment } M_{b2} = 19\,500 \text{ in.-lbs.}$$

Hence net effective moment is

$$M_b = 216\,000 - 19\,500 = 193\,500 \text{ in.-lbs.}$$

Since the section modulus remains the same, the maximum stress in the section y–y is

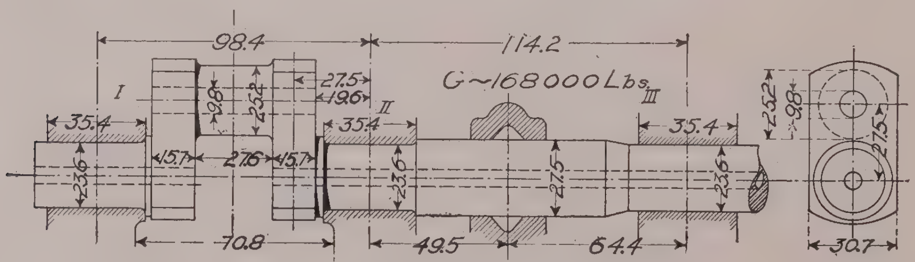
$$\sigma_b = \frac{196\,500}{16.1} = \mathbf{12\,200} \text{ lbs. per sq.in.,}$$

which is considerably greater than the stress in x–x.

Crank Position $\alpha = 40°$. For this the reactions due to T and Z are determined, and resolved into components parallel with and at right angles to the crank arm under discussion as above. The stresses resulting from these various forces are less than those shown above for the dead center position, as was proven by computation. For this reason the figures are not repeated here.

Main Journal. The computations for the main journal are made as previously outlined and offer nothing new.

Designs of Crank-shafts.



Figs. 191 and 192.—Dimensions in Inches.

Figs. 191 and 192. Crank-shaft of a single-acting blast-furnace gas engine. $D=52''$, $S=55.2''$, $n=90$ r.p.m. Weight of fly-wheel about 168 000 lbs., of the shaft, including counterweights, 48 400 lbs. Explosion pressure normally 25 atm. = 356 lbs. per sq.in.

The checking through of this shaft gives the following results: Maximum bearing pressure on the crank pin, $K_{\max}=1100$ lbs. per sq.in.; in each of the bearings I and II, $K_{\max}=460$ lbs. per sq.in. Maximum pressure in the bearings due to the weight of fly-wheel and shaft, in bearing I, $K=17.8$ lbs. per sq.in.; in bearing II, $K=135$ lbs. per sq.in., and in bearing III, $K=106$ lbs. per sq.in. The mean piston pressure, $P_m=106\,000$ lbs., is distributed as follows: on the crank pin, $K=\frac{106\,000}{25.2\times 27.6}=152$ lbs. per sq.in., in each of the bearings I and II, $K=63.2$ lbs. per sq.in. From these figures the average bearing pressures for one complete cycle are: for the crank pin $K=152$ lbs. per sq.in.; for bearing I, $K=17.8+63.2=81.0$ lbs. per sq.in.; for bearing II, $K=135+63.2=198.2$ lbs. per sq.in., and for the outboard bearing III, $K=106$ lbs. per sq.in. Peripheral speed of the crank pin is $v=9.84$ ft. per sec., of each of the three bearings, $v=7.8$ ft. per sec. Finally, the friction work for the various bearings is: for the crank pin $Kv=152\times 9.84=1485$ ft.-lbs. per sec.; for bearing I, $Kv=81\times 7.8=632$ ft.-lbs. per sec.; for bearing II, $Kv=198.2\times 7.8=1550$ ft.-lbs. per sec., and for bearing III, $Kv=106\times 7.8=826$ ft.-lbs. per sec.

The following table shows the stresses computed on the basis of the method developed on p. 164, etc.

Dead Center Position:

$\sigma_b=13\,100$ lbs. per sq.in., crank pin;

$\sigma_b=\ 9100$ lbs. per sq.in., second crank arm;

$\sigma_r=\ 6050$ lbs. per sq.in., second main journal.

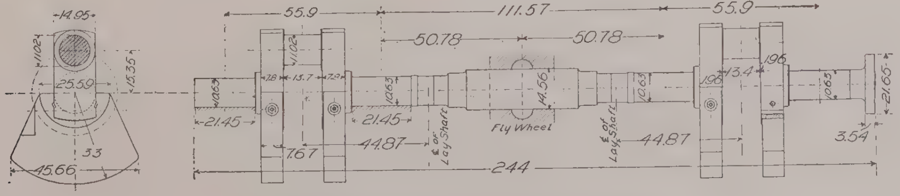
For the position of maximum turning effort ($\alpha=40°$):

$\sigma_r=\ 9800$ lbs. per sq.in., crank pin;

$\sigma_r=\ 5830$ lbs. per sq.in., second crank arm;

$\sigma_r=\ 7450$ lbs. per sq.in., second main journal.

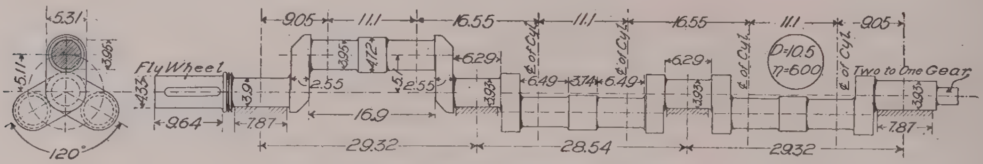
The maximum deflection of that part of the shaft carrying the wheel is in this case less than .0006″.



[Figs. 193 and 194.—Dimensions in Inches.

Figs. 193 and 194. Crank-shaft of a double-acting two-cylinder 4-cycle engine. $D=25.2''$, $S=30.7''$, and $n=150$ r.p.m. The assumed unit explosion pressure of 356 lbs. per sq.in. shows a maximum bearing pressure on the crank pin of 1160 lbs. per sq.in., and on the main journal of 384 lbs. per sq.in. Since the weight of fly-wheel is not given, a complete examination of this shaft cannot be made.

Figs. 195 and 196. Triple-throw crank-shaft of a six-cylinder single-acting marine engine (Loutzki). Normal rating 300 H.P. Assuming $p_z = 356$ lbs. per sq.in., we find the maximum bearing pressure for the crank pin = 1240 lbs. per sq.in.; for the inner main bearing it is 850 lbs. per sq.in. The velocity of rubbing is, in the one case, 10.8 ft. per sec.; in the other, 10.3 ft. per sec.



FIGS. 195 and 196.—Dimensions in Inches.

The design of the crank mechanism must finally take into account the question of **balancing.** This is especially important in the case of high-speed engines, or those for which sufficiently rigid foundations cannot be had. The forces to be balanced are (*a*) those due to the centrifugal effect of the crank arms and that part of the connecting-rod considered as a rotating mass, and (*b*) those due to the inertia pressure (see p. 156) of the reciprocating parts. The latter force acts in the direction of the center line of the cylinder. There is no difficulty encountered in exactly balancing the effect of the rotating weights (*a*) by other rotating weights having the same static moment. The reciprocating parts (*b*) of weight G_t pounds, applied at the crank radius r (see p. 156), require for perfect balancing a counterweight G_s, applied at an arm r_1, and diametrically opposed to G_t, such that $G_t r = G_s r_1$.

The counterweight G_s (assuming the connecting rod of infinite length) causes a centrifugal force equal to

$$C = \frac{.031\, G_s v^2}{r_1} = (.00034\, G_s r_1 n^2) \text{ lbs.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (66)$$

Of this force the component acting in the direction of the cylinder axis for any given crank angle α is

$$C_1 = C \cos \alpha = (.00034\, G_s r_1 n^2 \cos \alpha) \text{ lbs.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (67)$$

This force balances the inertia effect due to G_t. The second component of C, that at right angles to the axis of the cylinder, is

$$C_2 = C \sin \alpha = (.00034\, G_s r_1 n^2 \sin \alpha) \text{ lbs.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (68)$$

This force, C_2, in the case of horizontal machines, is taken up directly by the foundation, but in vertical machines its action across the frame may set up serious vibrations. For this reason it is usual in *stationary vertical* machines to balance only the rotating parts when any balancing is done at all. The axial inertia forces in vertical machines really compel the use of balance weights only when the effect of these forces approaches the weight of the engine. The weight of stationary vertical machines is usually from 100 to 150 lbs. per sq.in. of piston and that of high-speed auto-engines from 3 to 12 lbs. Hence the inertia forces are always less than the weight of the stationary vertical engines, but for the automobile engines the reverse is usually the case. Now since the body of automobiles is much more strongly affected by vertical than by horizontal forces as regards vibration, it pays, in the case of *vertical* auto-engines, to completely balance both the rotating and the reciprocating masses, irrespective of the action of the lateral force C_2.

Some designers of automobile engines use the following empirical formula, taken, as far as the author is aware, from French automobile practice·

$$G_s = P_{\frac{1}{3}} + \frac{P_{\frac{2}{3}} + K}{4} \text{ lbs.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (69)$$

In this formula $P_{\frac{1}{3}} + P_{\frac{2}{3}} = P =$ the weight of the connecting-rod, $K =$ the weight of the piston. $P_{\frac{1}{3}}$ is the weight of the crank pin end of the rod determined when the rod is supported on a knife-edge at a distance equal to one third of the rod length from the crank-pin end. $P_{\frac{2}{6}} = P - P_{\frac{1}{3}}$ is the remainder of the rod weight. It is said that vertical single-cylinder engines are nearly perfectly balanced if the counterweights are determined according to this formula.

Example. A six horse-power automobile engine has a cylinder diameter equal to 4.33″, stroke $= 4.73''$; $n = 1200$ r.p.m. The weight of the connecting rod $= P_{\frac{1}{3}} + P_{\frac{2}{3}} = 3.88$ lbs.; of the piston is 5.08 lbs. Supporting the rod, on narrow side, upon a knife-edge one third of the distance from the crank-pin center, and resting the crank-pin end on scales, the weight shown is $P_{\frac{1}{3}} = 2.16$ lbs. Hence $P_{\frac{2}{3}} = 3.88 - 2.16 = 1.72$ lbs. counterweight, if applied at an arm equal to the crank radius, should therefore weigh

$$G_s = 2.16 + \frac{1.72 + 5.08}{4} = 3.86 \text{ lbs.}$$

The correctness of this computation was checked very closely by supporting the engine shaft on knife-edges.

A theoretically perfect balancing of the inertia forces is rendered impossible by the finite length of the rod. This causes an unequal distribution of these forces over the two halves of the stroke, while the action of the transverse component due to the counterweight is always symmetrical. Taking the best case, i.e., when $G_t r = G_s r_1$, the amount of vibration in a machine balanced to this extent compared to that found in a machine entirely unbalanced may be expressed by the relation $\frac{r}{L} : \left(1 + \frac{r}{L}\right)$. With $\frac{r}{L} = \frac{1}{5}$ as an average case, there would then still remain a force equal to $\frac{\frac{1}{5}}{1\frac{1}{5}} C_a = \frac{1}{6} C_a$. If, on the other hand, the static moment of the counterweight is only $mG_t r$, in which of course $m < 1$, the above relation is rendered less favorable according to the ratio $\left(1 - m + \frac{r}{L}\right) : 1 + \frac{r}{L}$.[1]

If possible the counterweights should be placed in the planes of the crank arms. Counterweights in the fly-wheel cause additional twisting forces, because they are out of the plane of the rotating and reciprocating masses, and their use should be avoided, especially in case no outboard bearing is used. The room available between the main bearings is usually insufficient to admit of counterweights large enough to completely balance the reciprocating parts. It is usual therefore in vertical engines to balance only the crank arms and that part of the rod considered as rotating. In horizontal engines, also, the reciprocating parts are balanced as far as possible; in most cases, however, only about one half the weight necessary for this can be placed as indicated.[2]

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[1] Compare Radinger, Schnell-laufende Dampfmaschinen.

[2] See also p. 156.

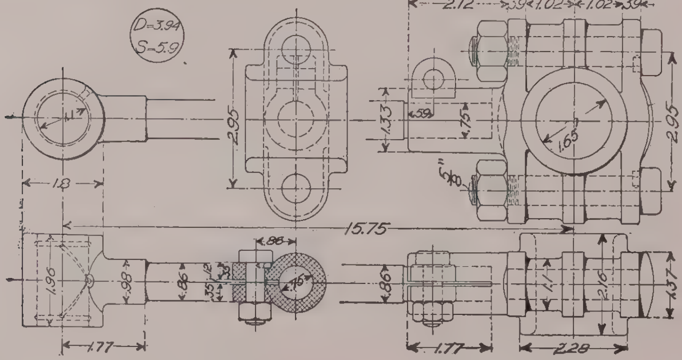
VI. Connecting=rods

Material. For the body of the rod usually soft steel; for the larger engines the use of cast steel is on the increase, while for the cheaper grade of small machines malleable iron is sometimes employed. Crank-pin bearings, when the pin diameter exceeds say 5″, are usually of cast iron or cast steel lined with white metal. For the crank-pin bearings of smaller engines and for the wrist-pin bearing the use of bronze is nearly universal.

In computations regarding the strength of the rod body considered as a long column, the usual factor of safety is $\mathfrak{S}=20$.

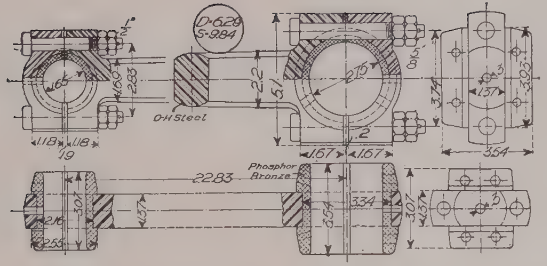
Crank-shafts for internal-combustion engines in most cases admit of two part crank-pin bearings only. Among these the so-called marine type of bearing seems to be the best adapted; the strap or stirrup type, on account of its lack of sufficient rigidity, should be used for the smaller sizes of engines only. For smaller rods the crank-pin bearing is also often made as a separate piece, the material being bronze, cast steel or cast iron (Figs. 197–200 and 205–206). One-piece construction of crank-pin bearing and rod body (see Figs 201–204 and 207–224), however, is to be preferred for the larger engines, because this arrangement assures greater stiffness and relieves the bolts of the inertia forces acting at right angles to the path of the rod. The wrist or piston-pin bearing is usually a solid head with bronze steps. White metal linings have usually not proven themselves well adapted for service in this end of the rod. Their application, however, is not entirely without promise of success, since of late years the piston-pin bearings of some of the large gas engines are lined with white metal and have so far given no trouble. In many of the smaller engines the wrist-pin bearing is designed without any arrangement for taking up wear, the bearing being simply a solid head with a bronze or hardened steel bushing. Rods with forked wrist-pin ends, Figs. 217–224, are used in connection with cross-heads only. In the larger sizes of trunk piston, the wear in the wrist-pin bearing should be taken up by drawing up that half of the bearing not under load, that is, the inner half. Owing to the lack of room, however, and the general inaccessibility of this bearing, this procedure is not easy with the usual methods of construction. Special constructions with this aim in view are shown in Figs. 225–229. In these the inner step may be drawn up by adjusting the bolt in front of the pin. Connecting rods with separate crank-pin heads may be so made that the rod length may be varied in order to change the compression, Figs. 197–200 or 205–206. This is of special advantage in experimental machines.

Design of Connecting Rods.



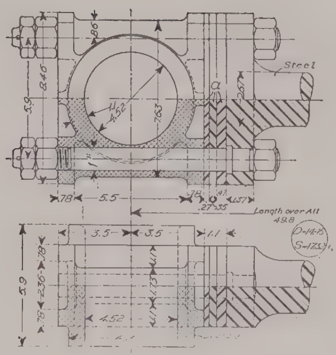
Figs. 197–200.—Connecting Rod for Güldner 2-cycle Engine, Fig. 35, p. 86.

(Body of rod cylindrical, as as it passes through a stuffing box. The brass crank-pin bearing is merely clamped on the rod body in order to be able to adjust the length of the rod or the size of the clearance space.)



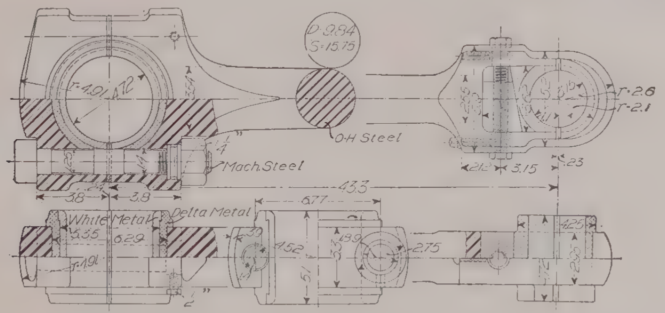
FIGS. 201–204.—Connecting Rod for a 6 H.P. Bánki Engine.

(With reference to mass-production and cost of manufacture, the rod body is made practically rectangular. The wide sides of the rod are either planed or milled, the narrow sides are milled, no lathe work being done.)



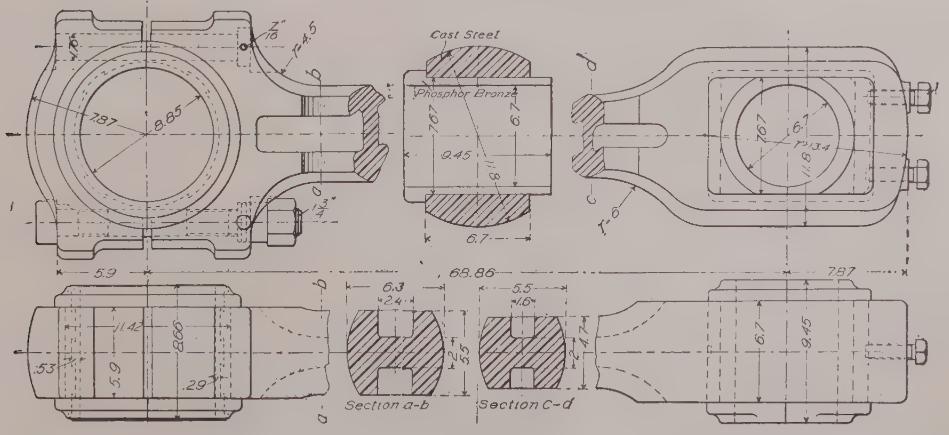
FIGS. 205 and 206.—Crank-pin End of Connecting Rod for a 40–50 H.P. Hornsby-Akroyd Engine.

(Plates *a* are used to adjust the length of the rod, and consequently also the compression space, to suit the kind of oil used.)



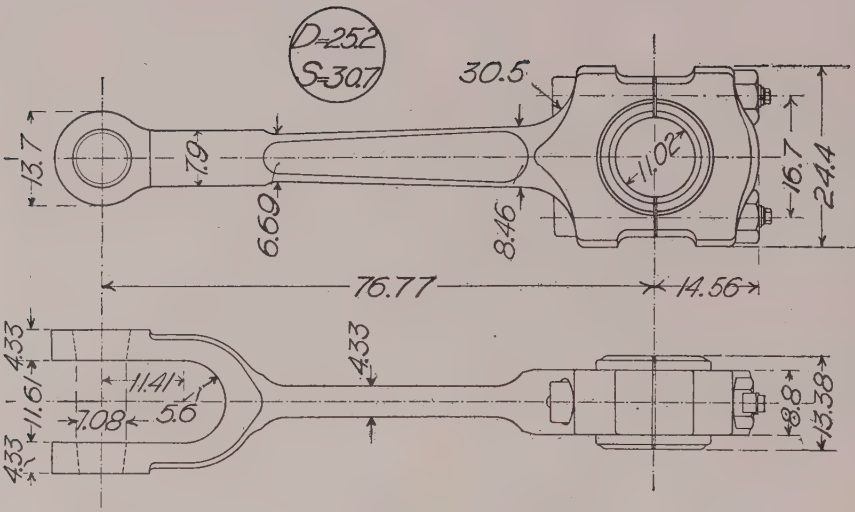
FIGS. 207–209.—Connecting Rod for a 20 H.P. Bánki Engine (using water injection). Built by Ganz & Co., Budapest.

(Standard construction for rods for all of the larger engines.)

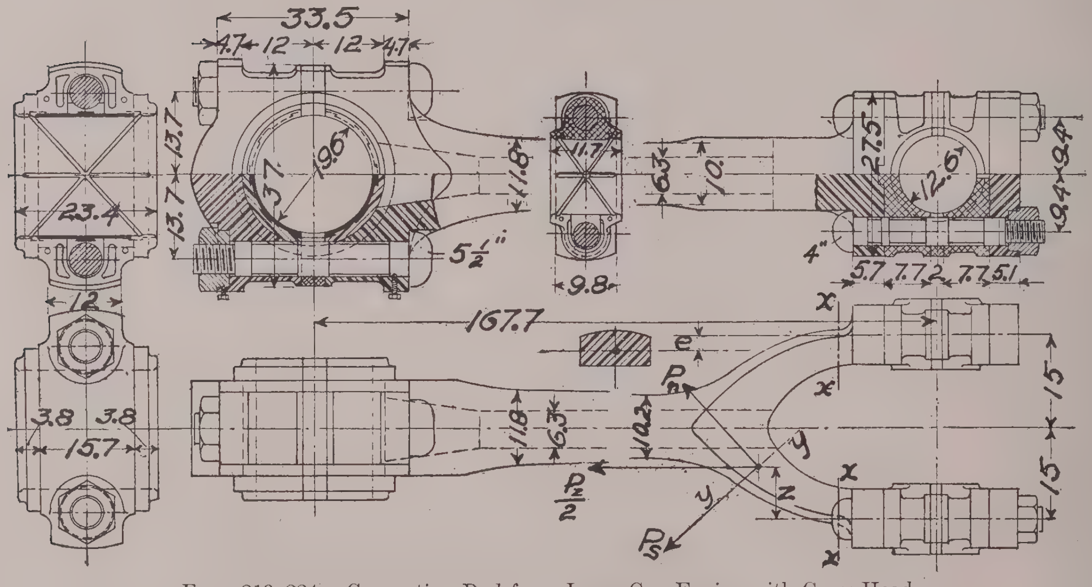


FIGS. 210–216.—Connecting Rod for 100 H.P. Güldner Engine ($D = 18.75''$, $S = 27.5''$).

(The rod body is cast steel, the crank-pin bearing cast iron lined with white metal, the wrist-pin bearing is bronze.)

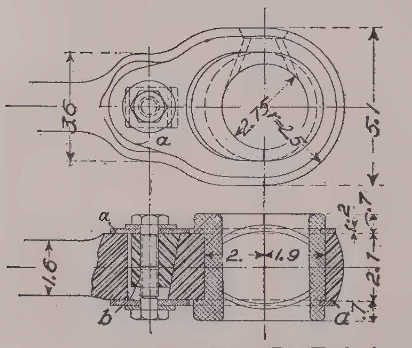


Figs. 217 and 218 —Connecting Rod, 300 H.P. Double-acting Otto Engine.



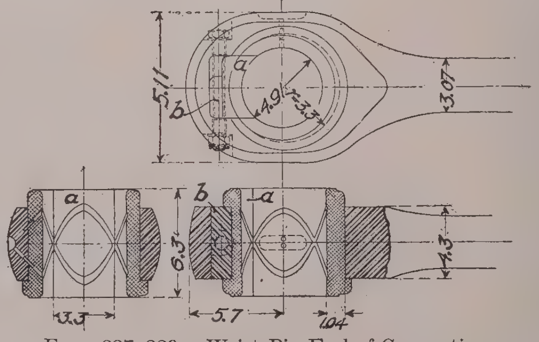
Figs. 219–224.—Connecting Rod for a Large Gas Engine with Cross Head.

(Concerning the forces $\frac{P_z}{2}$, P_s and P_n, see p 193.)



Figs. 225 and 226.—Wrist Pin End of Connecting Rod.

(The cover plates *a–a* surround the inner, unloaded half of the bearing, which may be drawn up by wedge *b*. Plates *a* are pressed firmly against the sides of the head by means of the wedge screws and are thus held in position.)



Figs. 227–229.—Wrist Pin End of Connecting Rod.

(The bearing is in two parts, block *a* and wedge *b* are located in that part of the bearing not under load.)

Constructive Details. The strength computations mainly concern themselves with designing the rod body as a long column safe against failure under the load P_z. Care should be taken, however, to see that the crushing stress in the smallest rod section does not exceed the safe limit. The bending stress resulting from the friction in the bearings and the weight of the rod may be neglected (see Example, p. 193). In large engines with high rotative speed, the inertia forces in the plane of the rod cause additional bending stresses which should be computed.

The general long-column formula of Euler (for a round-ended column) is

$$P_k=\pi^2\frac{EI}{L^2}\text{ lbs. (breaking load)} \quad \ldots\ldots\ldots\ldots \quad (1)$$

Putting $E=30000000$ lbs. per sq.in., the smallest moment of inertia of the rod body for the explosion load P_z must be

$$I=\frac{\mathfrak{S}P_zL^2}{300000000}\text{ in.}^4, \quad \ldots\ldots\ldots\ldots \quad (1a)$$

where $\mathfrak{S}$ = factor of safety.

Placing $\mathfrak{S}$ at 20, we have from eq. (1) the allowable load on the rod

$$P_z=\frac{\pi^2\times30000000\times I}{20L^2}\sim15000000\frac{I}{L^2}\text{ lbs.} \quad \ldots\ldots\ldots \quad (2)$$

For a circular cross-section

$$I=\frac{\pi d^4}{64}.$$

Hence, for a *round rod*, the required mean diameter of the body is

$$d_m=\sqrt[4]{\frac{64\,P_zL^2}{15000000\pi}}=\sqrt[4]{\frac{P_zL^2}{738000}}\text{ in.} \quad \ldots\ldots\ldots \quad (3)$$

From this, with $p_z=356$ lbs. per sq.in., and $P_z=280\,D^2$, for the standard diagram, we finally have

$$d_m=\sqrt[4]{\frac{D^2L^2}{2640}}=\sqrt{\frac{DL}{52}}\text{ in.} \quad \ldots\ldots\ldots\ldots \quad (3n)$$

Toward the wrist pin the diameter decreases to from .7 to .75 d_m; toward the crank pin it increases, depending upon the requirements of the crank-pin bearing; if necessary, the width required for the bearing at this end may be obtained by flattening the rod laterally (see Figs. 207–209).

For a *rectangular rod* of mean height h and width b, the moment of inertia (still considering the rod as a pin-ended column, that is, failure occurring in the plane of the rod) is

$$I=\tfrac{1}{12}\,h^3b.$$

Hence for this case

$$P_z=\frac{\pi^2\times30000000\times h^3b}{20\times12L^2}\sim\frac{1250000\,h^3b}{L^2}\text{ lbs.} \quad \ldots\ldots\ldots \quad (4)$$

The value of h is usually $= 1.7$ to $2\,b$. Substituting in (4), $h = 1.85b$, and solving for b,

$$b = \sqrt[4]{\frac{P_z L^2}{7950000}} \text{ in.}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

or, for the standard diagram,

$$b = \sqrt[4]{\frac{D^2 L^2}{28400}} = \sqrt{\frac{DL}{169}} \text{ in.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5n)$$

In the usual constructions b is kept constant throughout the length of the rod. The factor of safety used, $\mathfrak{S} = 20$, is a little smaller than is commonly employed in stationary steam-engine practice, but is quite safe. Even a further reduction of $\mathfrak{S}$ is permissible, since the maximum load P_z acts for a very short time only, which tends to prevent serious deformation. For the connecting-rods of small high-speed engines $\mathfrak{S}$ is sometimes decreased to 5, to keep the inertia forces of the rod down to the lowest practicable limit.

The maximum bending moment, in the plane of the motion of the rod, caused by the inertia forces is, according to Bach,

$$M_b = .000002 n^2 r A \gamma L^2 \text{ in.-lbs.}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

and the resulting stress therefore is

$$\sigma_b = .000002 n^2 r A \gamma \frac{L^2}{W} \text{ lbs. per sq.in.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

In these equations

n = revolutions per minute;
r = radius of crank in inches;
A = area of mean section of rod in square inches;
γ = weight of rod material in pounds per cubic inch; and
L = length of rod in inches.

Putting $W = \frac{1}{6} bh^2$ we shall have as a special equation applying to rectangular rods,

$$\sigma_b = .000012 n^2 r A \gamma \frac{L^2}{bh^2}$$

or since $A = bh$,

$$\sigma_b = .000012 n^2 r \gamma \frac{L^2}{h}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

Now the requirement is that

$$\sigma_b + \frac{P_z}{A} = \sigma_b + \sigma \leqq K_b,$$

hence, for the case of the rectangular rod,

$$K_b \geqq \frac{P_z}{bh} + .000012 n^2 r \gamma \frac{L^2}{h} \text{ lbs. per sq.in.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

The moment of inertia of the I section commonly employed for cast-steel rod bodies is $I=\frac{BH^3-bh^3}{12}$. In rods of this section the material is utilized to better advantage than is the case in round or rectangular rods, and these rods are consequently lighter. For cast steel, E may be taken at an average equal to 31 000 000.

Example. Engine dimensions are $D=37.5''$, $S=51.3''$, $n=125$ r.p.m. Here

$$P_z=280\times 37.5^2=396\,000 \text{ lbs.,} \quad \text{and} \quad L \text{ is taken at } 5r=5\times\frac{51.3}{2}=128''.$$

For a *round rod:*

$$d_m=\sqrt[4]{\frac{396\,000\times 128^2}{738\,000}}=9.7''\sim 9\tfrac{3}{4}''.$$

For a *rectangular rod,* mean thickness:

$$b=\sqrt[4]{\frac{396\,000\times 128^2}{7\,950\,000}}=5.35''\sim 5\tfrac{3}{8}''.$$

The mean height of the rod then is

$$h=1.85\times 5.37=9.93''\sim 10''.$$

In the case of the round rod the area of the mean section is 74.57 sq.in., for the rectangular rod it is 53.7 sq.in., which shows that for the same service the latter type of rod is somewhat lighter than the former. The inertia forces cause a bending stress in the rod of

$$\sigma_b=.000012\times 125^2\times 25.6\times .29\times\frac{128^2}{10}=2300 \text{ lbs. per sq.in.}$$

This stress is considerable, but the stress does not exceed the allowable, for since

$$\sigma=\frac{396\,000}{5.37\times 10}=7400 \text{ lbs. per sq.in.,} \qquad \sigma_{\max}=\sigma+\sigma_b=7400+2300=9\,700 \text{ lbs. per sq.in.} <K_b.$$

The forked wrist-pin ends of connecting rods, Figs. 217–224, receive their greatest stresses in the section x–x or y–y. In the first of these we have the crushing stress z

$$\sigma=\frac{.5\,P_z}{bh},$$

and the bending moment

$$M_b=.5\,P_z e.$$

In the section y–y we have the shearing force P_s, the twisting force P_n, and the bending moment $M_b=.5\,P_z z$. These separate stresses thus appearing are to be combined as previously shown.

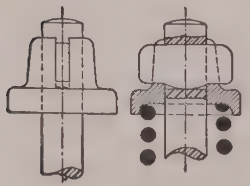
In *single-acting engines* the caps and bolts of the connecting-rod bearing are only under the load due to inertia of the reciprocating parts and the friction of piston and cross-head. This load in stationary engines does not usually exceed 10 lbs per sq.in. of piston. As far as the cap is concerned, therefore, the size required to hold the bearings is of more importance than the stresses it has to resist. It is also necessary to make the diameter of the bolts greater than this comparatively small load calls for, in order to compensate for any stress due to the strong drawing up required and to allow for the inertia effect of the rod in its plane of motion. Weak crank-pin cap bolts are a very serious danger during operation, and this particular machine part calls for ample dimensions as well as for care in the selection of material and in manufacture. (Soft O-H steel, no sharp grooves or corners, smallest bolt cross-section outside of thread, safety against working loose, etc.) In *double-acting engines* of course both the connecting-rod bolts and the crank-pin cap must be designed for the maximum piston load P_z.

VII. Valves

Material. For valve cages, or valve seats, hard close-grained cast iron; for valve disks mostly medium carbon steel or nickel steel; for small automatic valves sometimes also soft steel; large valves, especially those water-cooled, are made with cast-iron disks and stems of medium carbon steel.

The *allowable stress* in valve disks should be very low on account of high temperature, to guard against warping, and to furnish material for re-grinding. Numerical data is given in the computations to follow.

The valves of the working cylinders of gas engines of to-day are exclusively of the poppet type. The inlet valves may be automatic, but the exhaust valves must be mechanically operated. Slide valves, on account of the usual high explosion pressure and temperatures, can only be used in places where their dimensions are small and their stroke short, as ignition or starting valves, for instance. For any other purpose they are obsolete. For all important valves in regular and continuous operation, the vertical form of valve should be used. Horizontal valves are much more liable to leakage, sticking, and other interruptions of regular operation and are not reliable especially when automatic. If the use of the horizontal form is unavoidable, make the disk as light, the guide as long, and the lift of the valve as short as possible. And generally, in the case of horizontal valves, flat seats are better than the usual conical seat of poppet valves.


FIGS. 230 and 231. Fastening of Spring Plate.

Automatic inlet valves are applicable to small and comparatively slow-speed engines only. At the higher speeds the accompanying noise becomes very annoying, the wear is increased very seriously, and the volumetric efficiency is low. For these reasons mechanically operated inlet valves have of late years been used even in small auto-engines. In the larger machines the use of the automatic inlet valve is prohibited alike by decreased economy and capacity and by general unreliability in operation. The weak point of all automatic valves is rapid wear of the disk and liability to fracture. With this type of valve, screw fastenings are almost useless, and consequently the spring plates and other parts, in order to stay in place for any length of time, must either be pinned or keyed in (see Figs. 230 and 231), or they must be in one piece with the stem. The breaking away of the disk from the stem may, in the case of vertical engines whose valves are placed in the head directly over the piston, become positively dangerous, and safety appliances for the catching of loose disks should therefore be provided in such cases.

It is advantageous to place the exhaust valve disk in a water-cooled valve housing, or in the water-cooled cylinder head, so that the valve can be cooled by the jacket direct. When separate uncooled valve seats, as bushings or cages, are employed the proper cooling of the valve disk is much more difficult. In single-acting horizontal machines, the inlet valve should, if possible, be placed directly over the exhaust valve, because the incoming new mixture will help to keep the exhaust valve cool, and the inlet valve works easier if opening downward. Since the exhaust valve disk at the moment of opening the valve is loaded with from 30–60 lbs. per sq.in., the reaction upon the valve gear when large valves are used may be quite serious. To avoid this difficulty, double-seated valves have been devised which are partially balanced either by the pressure of the exhaust gases themselves or by compressed air furnished for the purpose. To gain the same end, two valves instead of one are sometimes employed. Another scheme is to use one valve and an auxiliary exhaust port. When two valves are used, one should open a little ahead of the second for the purpose of relieving it. For the same reason, in case the auxiliary port is used, the piston should start to uncover this before the exhaust valve begins to open. By means of this preliminary exhaust the exhaust gases of highest pressure and temperature are discharged in a very satisfactory way, leaving nothing but low pressure, and hence comparatively cool gas, for the exhaust valve to handle.

The exhaust gases, owing to their high temperature, have a very serious effect upon the valve disk, proper lubrication is very difficult, and hence above a certain size, say 100 H.P., direct water cooling of the disk must be resorted to. (For example see p. 199). It has been attempted to combine the inlet and exhaust valves, and thus not only to keep the combined valve comparatively cool by the incoming fresh charge, but also to simplify the mechanical details of the machine; but none of these forms, of which Figs. 250 and 251 give some examples, have been able to maintain themselves. If they are to be at all reliable, their construction is not cheaper but more costly than that of two simple valves, especially when the inlet is mechanically operated. The sucking back of exhaust gases from the valve housing can hardly be prevented, and the double valve is too vulnerable against soot, rust, etc. Such three-way valves, invented and exploited time and again therefore appear to the experienced designer to offer little promise of utility.

The main aim of *gas* and *mixing* valves is to produce as far as possible a uniform mixture of gas and air, and this determines the general features of their design. The means employed to this end are: Increasing the opportunity for diffusion by dividing the stream of gas into many fine streams or broad thin layers, proper guiding of the various gas currents, mechanical agitation, etc. In the case of automatic mixing valves, care should be taken to see that for all valve positions the ratio of the areas of gas and air inlet-ports shall be the same, otherwise the composition of the mixture depends upon the piston speed and the frictional resistance of the valves. Positively operated mixing valves in general give a better guarantee of constant mixture. For gases not entirely free from impurities (suction gas), automatic mixing valves are not satisfactory in any case.

The starting valves, found in all the larger engines, operate only for a few turns of the engine at a time, and hence its construction is not of very great importance. However, to avoid any trouble due to this inactivity, the valve should be so placed that it will keep fairly cool and that no chance be given for sticking it by deposits of burned oil, etc. This valve may be actuated entirely by hand or temporarily and positively by a part of the valve gear. It is self-evident that the main valves should all open inward, and the same is true of the starting valves, even if this is constantly loaded by compressed air under a pressure greater than the maximum working pressure p_z in the cylinder. A common form of starting valve shows a design something like that of an ordinary globe valve, that is, the disk is screwed down on its seat. This form is not recommended, because the explosion pressure comes upon the rather weak valve stem and easily causes the valve to leak. Besides this, the contact surfaces are easily corroded and encrusted.

Design of Valves.

For forms of ignition valves, see p. 293, for starting valve, p. 251.

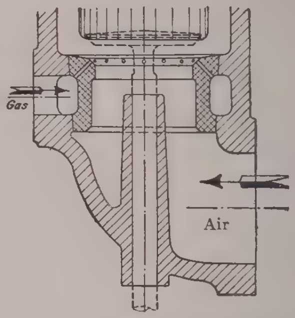


Fig. 232.—Inlet and Mixing Valve for Small Engines.

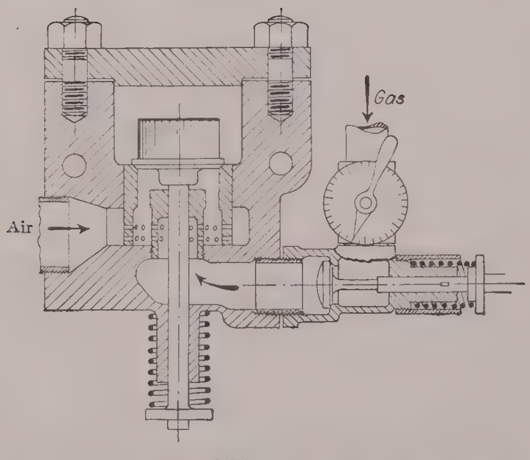


Fig. 233.—Inlet and Mixing Valve combined with Governing Valve, Brombacher Type.

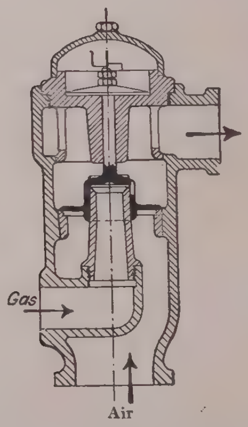
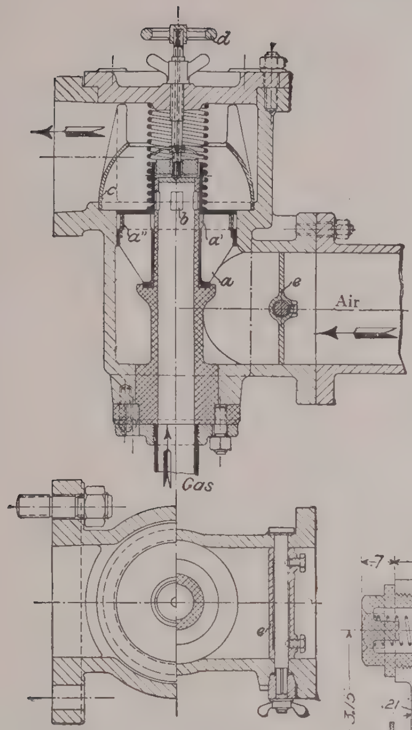


Fig. 234.—Mixing Valve, Koerting Bros. (Automatic disk valve with dash-pot at top).



FIGS. 235 and 236.—Mixing Valve, Güldner, used mainly on the large vertical engines.

(Disk *a* controls the gas ports *b* by means of the ports *a′*, while ports *a″* regulate the air supply. Bell *c* helps to make the mixture uniform. The lift of the valve is adjusted through *d*. Throttle valve *e* controls the air supply.)

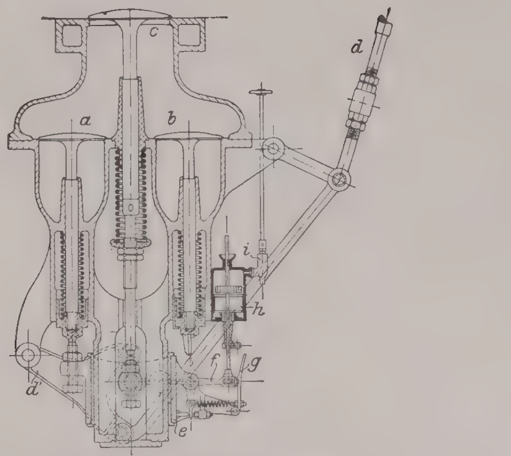
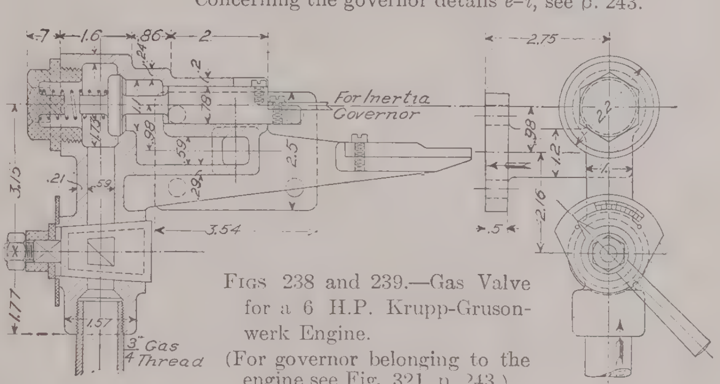


FIG. 237.—Inlet and Mixing Valve for a large Simplex Engine. (*a* air, *b* gas, *c* inlet valve, all operated by *d d′*.) Concerning the governor details *e–i*, see p. 243.



FIGS 238 and 239.—Gas Valve for a 6 H.P. Krupp-Grusonwerk Engine. (For governor belonging to the engine see Fig. 321, p. 243.)

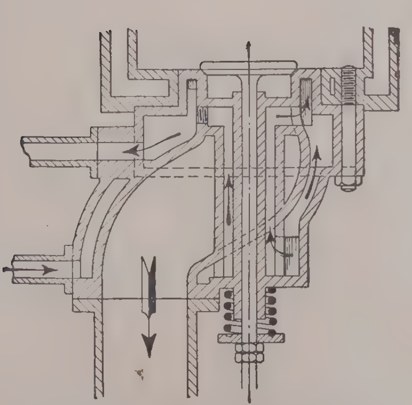
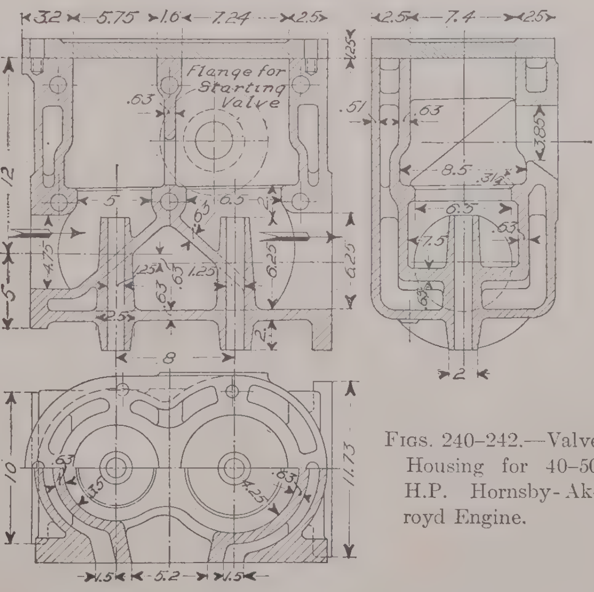


FIG. 243.—Exhaust Valve with Water-cooled Stem-Guide, used by Maschinenbau-Ges. Nürnberg for Medium Sized Engine.



FIGS. 240–242.—Valve Housing for 40–50 H.P. Hornsby-Akroyd Engine.

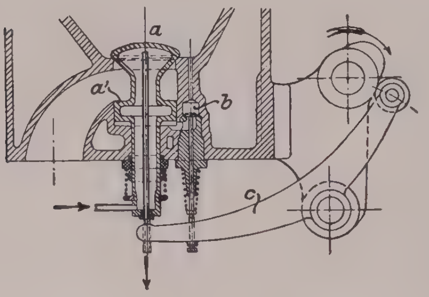
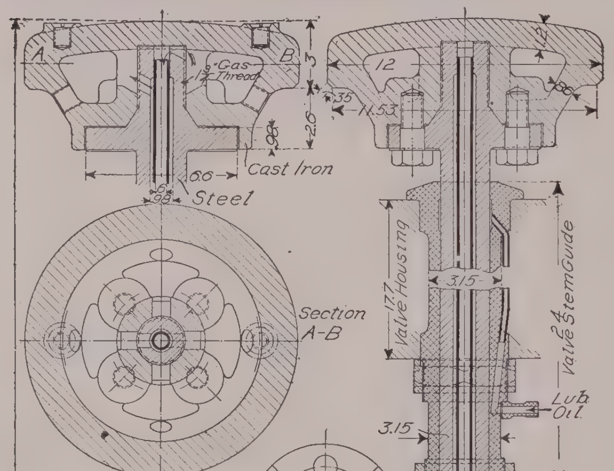


FIG. 244.—Water-cooled and balanced Exhaust Valve for a large Crossley Engine.

(*a* main valve, *b* auxiliary valve. This valve is operated by *c* and opens just ahead of the main valve. This allows the pressure in the cylinder to act on the under side of *a'* and thus relieves the main valve.)



FIGS. 245–249.—Water-cooled Exhaust Valve, showing Spring Suspension and Guide Construction, 600 H.P. Nürnberg Blast Furnace Gas Engine.

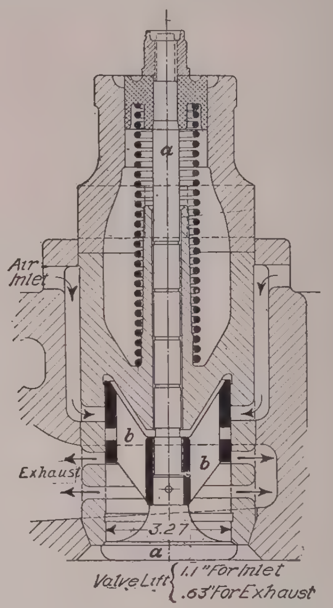


FIG. 250.—Combined Inlet and Exhaust Valve.

(The stem of the valve *a* carries a concentric sleeve *b*. The latter, for maximum lift, closes off the upper inlet ports, while for the smaller lift it opens the exhaust ports.)

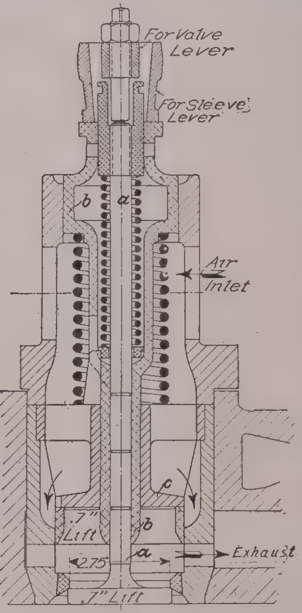


FIG. 251.—Combined Inlet and Exhaust Valve.

(Valve *a* and concentric sleeve valve *b* are operated independent of each other. During suction the valve *b* has, in its lowest position, closed off the exhaust ports, while during exhaust, *b* in its highest position, resting against *c*, has closed off the inlet ports.)

A valve of similar construction is shown in Part III in connection with the Loutzki automobile engine.

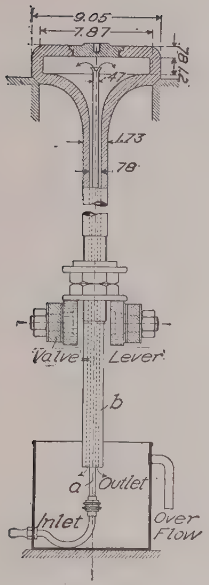


FIG. 252.—Exhaust Valve, Pawlikowsky type.

(Water-supply pipe *a* is stationary and surrounded by the valve spindle. The latter is bored out large enough so that the hot water can find its way out through the annular space between the inside wall of spindle and the water-supply pipes.

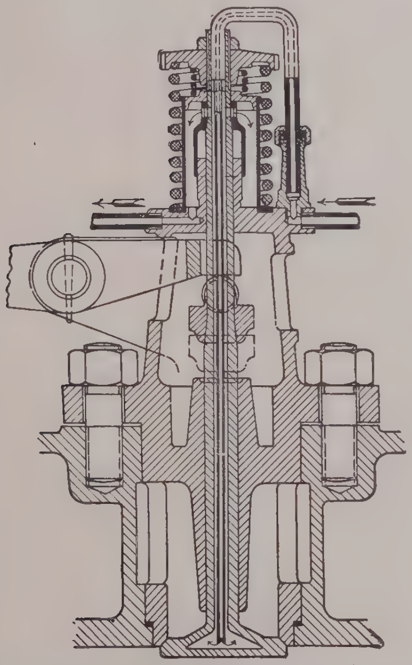


FIG. 253.—Exhaust Valve, Güldner Type with Pawlikowsky Method of Water Supply.

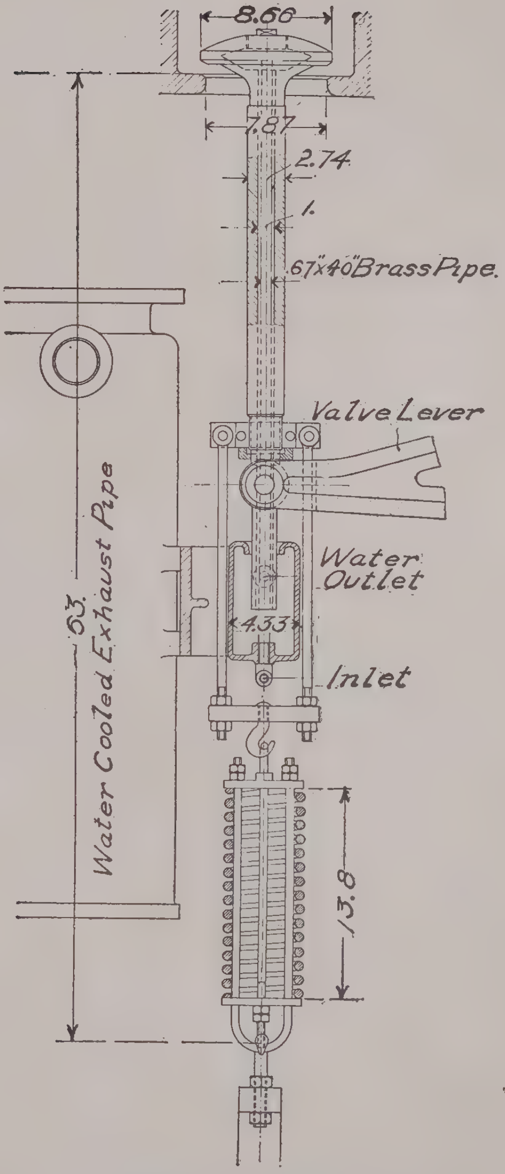


FIG. 254.—Exhaust Valve, Pawlikowsky Type, for 100 H.P. Körting Engine.

Constructive Details. Let C_s be the volume in cubic feet of gas to pass through a given valve per second, v the velocity of the gas in feet per second, h the lift of the valve , and d the diameter of passage, both in feet. Then the free cross-section f', required through the valve, in square feet, will be

$$f' = \pi dh = \frac{C_s}{v} \text{ sq.ft.}, \quad \ldots \ldots \quad (1)$$

if the lift in every case is $h \leqq \frac{d}{4}$. For the maximum effective lift $h = \frac{d}{4}$, we may write

$$C_s = .785 d^2 v \text{ cu.ft.}, \quad \ldots \ldots \quad (2)$$

and from this the required minimum diameter of the valve passage will be

$$d \geqq \sqrt{\frac{C_s}{.785v}} = \sqrt{\frac{1.274C_s}{v}} \text{ feet.} \quad \ldots \ldots \quad (3)$$

But in operation the introduction of a charge through a valve occurs intermittently with varying velocity, hence we are compelled, in determining the free cross-section required, to base our computations on the true interval of and the quantity of gas necessary for a single stroke. Assume that a volume of gas equal to the entire stroke volume $= V_h$ cu.ft. (without reference to the volumetric efficiency η_e) is to pass through the valve and let the diameter of piston be D'', the area of piston be F sq.in., S the stroke in feet, and c feet per second be the mean piston speed at n revolutions per minute. Further, let the clear diameter of the valve seat be d'', the lift of the valve be h'', and f sq.in. be the free cross-section neglecting the area of the stem. Then if v ft. per sec. is to be the mean velocity of the gases passing the valve, we must have

$$f = \frac{144V_h}{v} = \frac{Fc}{v} \text{ sq.in.} \quad \ldots \ldots \quad (4)$$

And, again, providing that h is always $\leqq \frac{d}{4}$, we may write eq. (4), since $c = \frac{Sn}{30}$,

$$f = \pi dh = \frac{\pi D^2}{4} \cdot \frac{Sn}{30v}, \quad \text{or} \quad dh = \frac{D^2 Sn}{120v} \text{ sq.in.} \quad \ldots \ldots \quad (4a)$$

The mean charging and discharging velocity v should not exceed 75 ft. per sec. (in large engines this may of necessity go as high as 100 ft. per sec.) and substituting this figure in eq. (4*a*) we finally have

$$dh = \frac{D^2 Sn}{9000} \text{ sq.in.} \quad \ldots \ldots \quad (5)$$

According to page 156, a ratio of connecting-rod to crank equal to 5 makes $c_{\max} = 1.6\,c$, so that if v is taken at a mean of 75 ft. per sec., the maximum velocity of passage through the valve will be 120 ft. per sec. To keep within this maximum velocity limit, any given piston position requires a valve lift of.

$$h \geqq \frac{D^2 Sn}{9000d} \psi \text{ in.} \quad \ldots \ldots \quad (6)$$

In this equation the factor $\psi = \sin \alpha \,(1 \pm \lambda \cos \alpha)$ expresses the variation of piston velocity when the connecting-rod ratio $\frac{r}{L} = \lambda$. For the usual ratio, $L = 5\,r$, Table 18 gives the numerical values of ψ.

TABLE 18

VARIATION OF PISTON VELOCITY (ψ) FOR $\lambda = 1:5$

Part of outstroke	.02	.04	.06	.08	.10	.15	.20	.25	.30	.35	.40	.45	.50
Part of instroke	.98	.96	.94	.92	.90	.85	.80	.75	.70	.65	.60	.55	.50
$\psi = \sin \alpha(1 \pm \lambda \cos \alpha)$	.304	.428	.516	.586	.648	.765	.853	.914	.962	.993	1.011	1.018	1.014
Part of outstroke	.55	.60	.65	.70	.75	.80	.85	.90	.92	.94	.96	.98	1.00
Part of instroke	.45	.40	.35	.30	.25	.20	.15	.10	.08	.06	.04	.02	0
$\psi = \sin \alpha(1 \pm \lambda \cos \alpha)$	1.00	.976	.939	.892	.832	.759	.668	.554	.496	.433	.355	.251	0

Plotting the values of ψ from the above table as ordinates and the corresponding piston positions as abscissæ, we shall obtain a sine curve similar to the piston velocity curve, Fig. 174. p. 155. From this curve we can at once determine the necessary valve lift and the outline of the actuating cam. If proper co-ordinates are chosen, the h curve, Fig. 255, will lie outside of the ψ curve throughout, but this can only be done when the valve starts to open slightly before the beginning of the stroke, and closes slightly after the end.

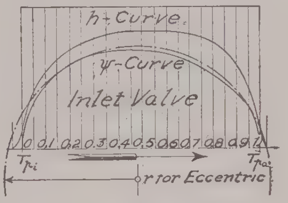


FIG. 255.

The maximum lift $\frac{d}{4}$ can be adequately utilized only in the smaller moderate speed machines. In large engines questions of valve design, and in high-speed engines the proper instant of closing of the inlet valve make it necessary to keep the maximum lift much below $\frac{d}{4}$. Automobile engines usually show $h = \frac{d}{10}$ to $\frac{d}{15}$ for automatic inlet valves.

If there should be more than one valve, each is to be designed as above indicated for the amount of gas it is called upon to handle. In hit-and-miss engines care should be taken to see that the valve held open during the misses has sufficient free opening in this position.

The dimensions of the valve cages, guides, or retainers, if they are used, depend upon the general conditions of the particular design. To make gas-tight joints between these parts and the cylinder heads or cylinder walls into which they are fitted, metallic packing rings or surfaces ground conical at an angle of 60° should be employed. If the cages are fairly large the force with which they must be pressed home to secure a tight joint is considerable, and if the conical joint is employed there is therefore danger of splitting the housing. In such cases, therefore, it is better to use flat surfaces with metallic packing rings.

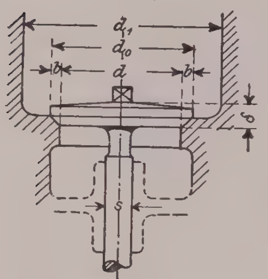


FIG. 256.

As far as the dimensions of the valve disk are concerned, besides d or d_0 (Fig. 256) only δ can be determined by computation. For mechanically-operated valves with disks of medium carbon steel we may make

$$\delta=\sqrt{\frac{p_2(.5d_0)^2}{6400}}\text{ in.,} \quad \ldots \ldots \ldots \ldots \quad (7)$$

or putting $p_2=356$ lbs.,

$$\delta=\sqrt{\frac{(.5d_0)^2}{18}}\text{ in.} \quad \ldots \ldots \ldots \ldots \quad (7n)$$

To derive this equation the disk has been considered a flat plate supported at the edge, and in view of the high temperature and the fact that an allowance should be made for regrinding, the allowable stress has been taken as $K_b=5700$ lbs. per sq.in. For the purpose of decreasing the weight of an automatic valve, K_b may be increased up to 11000 lbs. per sq.in. In the larger disks, say $d \geqq 4''$, δ may be decreased to from $\frac{4}{5}$ to $\frac{2}{3}\delta$ toward the edge.

The width b of the valve seat may be made approximately equal to

$$b=.5(d_0-d)=.01d+\tfrac{3}{16}\text{ in.} \quad \ldots \ldots \ldots \ldots \quad (8)$$

For automatic valves this should be made a little greater. The angle of the seat is in most cases about 45°.

For the diameter of the valve stem, the following formula gives satisfactory figures:

$$s=\tfrac{1}{8}d+\tfrac{1}{4}'' \text{ to } \tfrac{1}{8}d+\tfrac{7}{16}\text{ in.} \quad \ldots \ldots \ldots \ldots \quad (9)$$

The stem of the exhaust valve may be made a little greater in diameter than this in order to decrease the wear and to facilitate conduction of heat to the cooling water. The length of the guide depends primarily upon the general design of the valve cage or housing, and does not depend directly upon the size of the valve. It is evident that short valve actuating levers and considerable frictional resistances require longer guides for the valve stem than if the reverse were the case. To prevent possible jamming or tipping up of the valve disk, the guide should be extended if possible to the point where the actuating lever takes hold. But even this is not sufficient for the exhaust valves of large engines, for which a separate guide to take up the lateral pressure of the valve lever should be provided (see Fig. 252). Other points which require careful design in connection with large exhaust valves are the cooling and lubricating arrangements for the valve stem guides.

The diameter d_2 of the valve housing may be found from the requirement that

$$.785(d_1^2-d_0^2)>.785d^2. \quad \ldots \ldots \ldots \ldots \quad (10)$$

This is satisfied as long as

$$d_1 \geqq 1.6d. \quad \ldots \ldots \ldots \ldots \quad (11)$$

It is generally the case, however, that the form of the combustion chamber furnishes more than ample area in the annular space between disk and housing.

The valve springs are generally cylindrical, spiral steel springs. For these some general information will be given at the end of Part II, page 312. The requisite tension in these springs depends mainly upon the vacuum occurring in the cylinder. For mechanically-operated valves, the effect of this vacuum may be taken equal to a load of from 6 to 7.5 lbs. per sq.in. of cross-section of valve; for automatic valves these figures change to from 11.5 to 14.0 lbs. per sq.in. The tension of the springs should be such as to be able to overcome the suction action of the cylinder under all conditions of operation, to prevent accidental sucking back of air or exhaust gases. If the valve springs should also be employed to help fetch part of the valve gearing back into certain positions, the inertia of such parts of course calls for a corresponding increase in the power of the spring (see p. 205). It might be said that this method of actuating part of the valve gear through part of its movement is no longer good practice in the larger machines.

In the larger sizes of engines not only the load on the springs, but also their deflection in operation, are considerable, and for that reason care should be taken that the spring is not made too short. In the position of maximum deflection, that is, with the valve wide open, there should be from .05 to .10″ play between the individual coils. The number of coils and the length of the spring when free should be computed with this allowance. The following example shows the method:

Example. A given inlet-valve, opening downward, has a diameter in the clear of 4.73″ and a lift of 1.18″. The weight of the disk including spring plate, nut, and stem, is 14.5 lbs., which corresponds to a load of .85 lbs. per sq.in of passage. Maximum vacuum in the cylinder is 7.1 lbs., hence with closed valve the minimum tension in the spring should be $P_y = .785 \times 4.73^2 \times (7.1 + .85) = 139$ lbs.[1] When the valve is wide open, the tension P_0 will be from one third to one half greater than P_g. Now turning to Table 39, p. 312 the spring which seems to fit both the load and the disk best is one with a diameter of $2\frac{3}{4}$″ (measured as usual from center of wire) and with thickness of wire $(\partial) = .218$″. For this spring the table gives the maximum load $P_{\text{max}} = 185$ lbs., and a maximum deflection f, for every turn, equal to 1.08″. If a spring of $i = 10$ turns be used the maximum available deflection will therefore be 10.8″. Now, from a diagram, Fig. 257, drawn with P_{max} as abscissæ and f_{max} as ordinates, it is at once seen that in order to obtain a spring tension of 139 lbs., the spring must be compressed an amount equal to $z = 8$″. This must be the initial deflection of the spring when placed on the valve stem. The diagram also shows that by compressing the spring equal to the required lift $h = 1.18$″, the maximum tension in the spring will be $P_0 = 162$ lbs. (which satisfies the requirement that $P_0 \leq P_{\text{max}}$). The theoretical length of the free spring should be at least

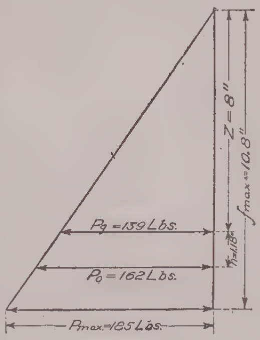


FIG. 257.

$$l' = i\partial + z + h = (10 \times .218) + 8.0 + 1.18 = 11.36 \text{ ins.}$$

With this length, however, and the valve open equal to the lift, the coils of the spring would be in contact, which is not permissible. To make some allowance for increase of initial compression of spring, should this be found necessary, the clearance between the coils should be made ample, say $e=\frac{\partial}{3}$. The required length of the spring then will be

---

[1] Where the computations must be extremely accurate, the effective disk diameter, i.e., up to the middle of the seat, should be used. In Fig. 256 this is $d + b$ instead of d.

$$l=i\delta+z+h+i\frac{\partial}{3}=(10\times.218)+8.0+1.18+\left(10\times\frac{.218}{3}\right)=12.08,\text{ say 12 ins.}^{1}$$

The accurate determination of the tension of valve springs is especially important in the case of the automatic inlet valves of high-speed engines, since this tension determines the suction resistance and the volumetric efficiency η_ε (p. 31) of the cylinder. Our present knowledge of the mode of valve action does not enable us to derive a rigid mathematical determination of the most favorable load on the valve disk. In view of this, the most rational method is to regard the valve disk as a body of weight G, or mass $m=\frac{G}{32.2}$, which must be accelerated during the time t, and through the distance h(=valve lift). The general equation for this case is

$$Ph=\frac{1}{2}mv^2=\frac{1}{2}\,\frac{G}{32.2}v^2, \quad \text{or} \quad Pt=mv=\frac{G}{32.2}v. \quad . \ . \ . \ . \ . \ . \ . \quad (12)$$

Putting $t=\frac{v}{32.2}=\sqrt{\frac{2h}{32.2}}$, we find the mean spring tension required in automatic inlet valves, for the time and lift chosen, is

$$P_f=\frac{2mh}{t^2}=\frac{2Gh}{32.2t^2}\sim\frac{Gh}{16t^2}\text{ lbs.}, \quad . \ . \ . \ . \ . \ . \ . \ . \ . \ . \quad (13)$$

and inversely,

$$t=\sqrt{\frac{Gh}{16P_f}}=.25\sqrt{\frac{Gh}{P_f}}\text{ sec.} \quad . \ . \ . \ . \ . \ . \ . \ . \ . \ . \ . \quad (14)$$

Equation (13) gives the required average minimum tension in the spring. Friction has here been left out of account. In the case of valves opening downward it may be assumed that any guide friction is balanced by the weight of the disk.

VIII. Valve Gearing

Material. For cam or lay shafts, machinery steel, cams and rollers, hardened steel; for the valve levers soft steel or cast steel; for driving gears generally, steel upon cast iron or upon bronze in the case of screw gears, and cast iron upon cast iron for spur (bevel) gears. For low intensity of pressure and low velocity, satisfactory service is also given by cast-iron worm gears, if constant lubrication is provided.

The allowable stress in the material may be taken as heretofore, provided a satisfactory determination of stresses can be made. If this is not possible we must fall back on experience and judgment.

---

[1] This computation assumes a maximum fiber stress of 125000 lbs. per sq.in , but the maximum load, from the diagram, is only 165 instead of 185 lbs., so that the real maximum fiber stress is only $\frac{165}{185}\times125000=111000$ lbs. See note accompanying Table 39.

The drawings in Part III give a general idea of the valve gearing usually employed for internal-combustion engines. They show that in horizontal engines the valves are generally actuated by cams or eccentrics situated on a lay shaft, which is driven at half speed by gearing from the crank shaft and passes usually along what may be considered the front side of the engine. The valve gearing of vertical machines does not show such standard type of design. In the smaller size of this machine, with exhaust valves at the side of the combustion chamber and automatic inlet valves, it is comparatively easy to actuate the exhaust valves by the use of a single lay shaft parallel to the crank-shaft. If, on the other hand, the valves are placed in the cylinder head, the construction grows a little more complicated, since the use of a single vertical lay shaft does not give a good chance for a generally satisfactory design. In such cases an intermediate shaft transmitting the motion from the main shaft to the gear shaft horizontally mounted above is usually required. This intermediate shaft with its two gears (Figs. 258 and 259) makes the engine cost more, although compared with the generally very low cost of construction of vertical machines, the increase caused thereby is of small importance. The construction shown in Figs. 261 and 262 meets all requirements for vertical engines up to say 20 H.P. The annular gear used works very quietly and accurately, and on account of its large contact surface is very well suited for high speeds.

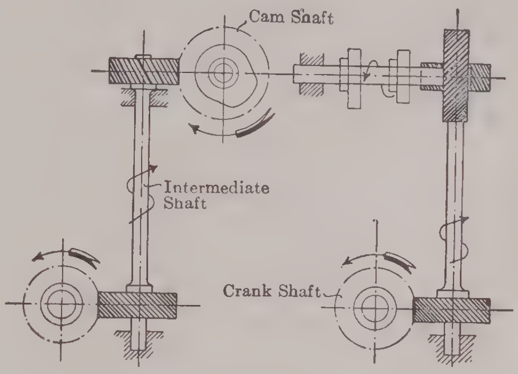


Fig. 258. Fig. 259.

Based on experience it may be said that, for the transmission of motion from the main to the lay shaft, a pair of spur or screw gears is best and most satisfactory. Many attempts have been made, because of lower cost, to use for this purpose various types of clutches, chain drives, etc., but none of the substitutes so far have shown any considerable measure of success. Attempts have also been made to replace the cams so commonly employed for actuating the valves by other devices of equal value. Cams do the work required of them very satisfactorily. The only other construction to be considered would be the eccentric. Eccentrics in themselves, however, are not very well suited to gas-engine requirements, and when used in the earlier types of small engines have been able to maintain themselves only in isolated instances. They cost more than cams and are less simple, offer less range of adjustment as far as the motion of the valve is concerned and possess especially the disadvantage that the lift of the valve is comparatively slow. To avoid the latter drawback, either large eccentricity, which means large eccentric dimensions, must be used, or recourse must be had to complex valve gearing with its accompanying lost motion. Fig. 255 shows that in such a case only about one third of the motion of the eccentric is utilized.

It cannot be denied, of course, that cams also have serious disadvantages when used in engines above a certain capacity. The pressures between cam and roller at the beginning and end of the motion, and their reaction on the rest of the valve gear, finally grows very heavy, which becomes manifest in noisy operation, rapid wear of contact surfaces, etc. This fact has, in the design of large engines, led to the adoption of the eccentric valve drive in combination with a peculiar system of rocking valve levers by which the closing of the valve is mechanically controlled. (See p. 210, Figs. 278–282.) Other manufacturers, imitating the well-known valve gear construction of steam engines, have equipped their large engines with simple eccentric motions and attempted to get a satisfactory valve action, in combination with so-called wiper cams or similar devices. Whether these innovations will lead to any general use of the eccentric in the design of large engines will depend largely upon the practical results obtained. Information on this point is as yet quite meager. The satisfactory service of this type of valve in steam-engine practice can hardly be accepted as a criterion because the conditions of operation there are quite different (almost completely balanced valves, considerably smaller lifts, shorter time of valve action, etc.).

Designs of Valve Gearing.

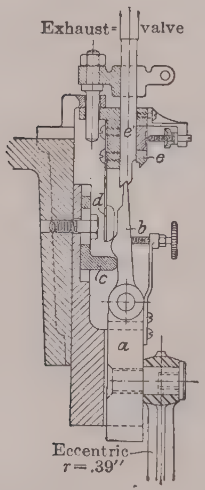
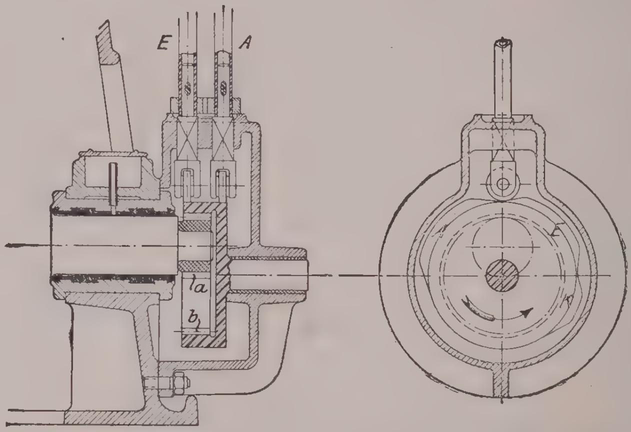


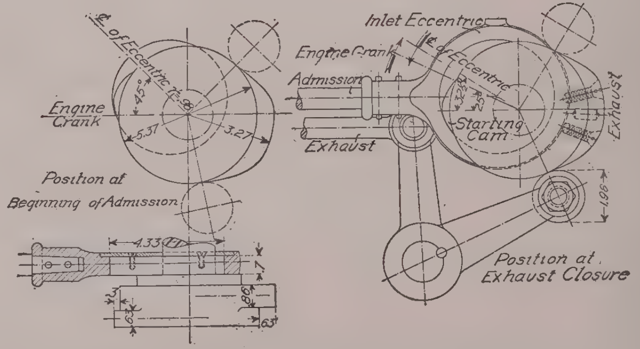
Fig. 260.—Combined 4-cycle Valve Gear and Pendulum Governor, G. Wenzel.

(Slide a and pick-blade b are operated from the crank shaft. Sliding piece e with deflector d is held to e' only by friction and follows the motion of e' to the lowest position. In this position, d will deflect the ascending blade b, so that it will miss the stem e', but will strike e, moving this upward along e'. At the next upstroke however b will strike e' before it encounters d. If the speed is above normal, b will be constantly thrown out a sufficient amount to miss e' by the deflector c.)

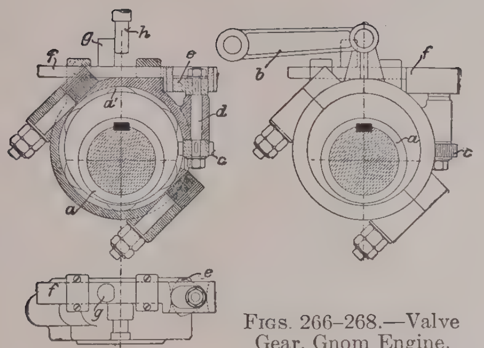


Figs. 261 and 262.—Valve Gear for an 8 H.P. Güldner Engine.

(The spur gear a drives the annular gear b, which, on its circumference, carries the inlet, exhaust and compression-relief cams E, A, and K.)



Figs. 263–265.—Inlet Valve Gear Employing Eccentric, 6 H.P. Engine, Krupp-Grusonwerk. (Exhaust is operated by cam.)



Figs. 266–268.—Valve Gear, Gnom Engine.

(Eccentric *a*, whose motion is guided through lever *b*, rotates the small gear *c* by means of the screw threads *d'*. Cam *e*, on the same shaft with *c*, operates the slide *f*, so that every other turn of the engine the blade *g* strikes the exhaust valve gear *h*.)

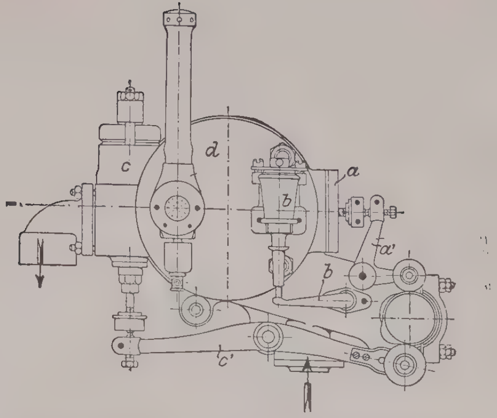
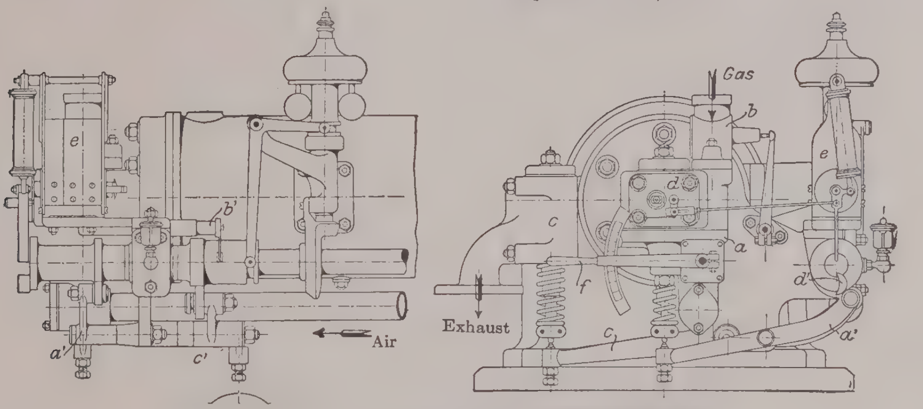


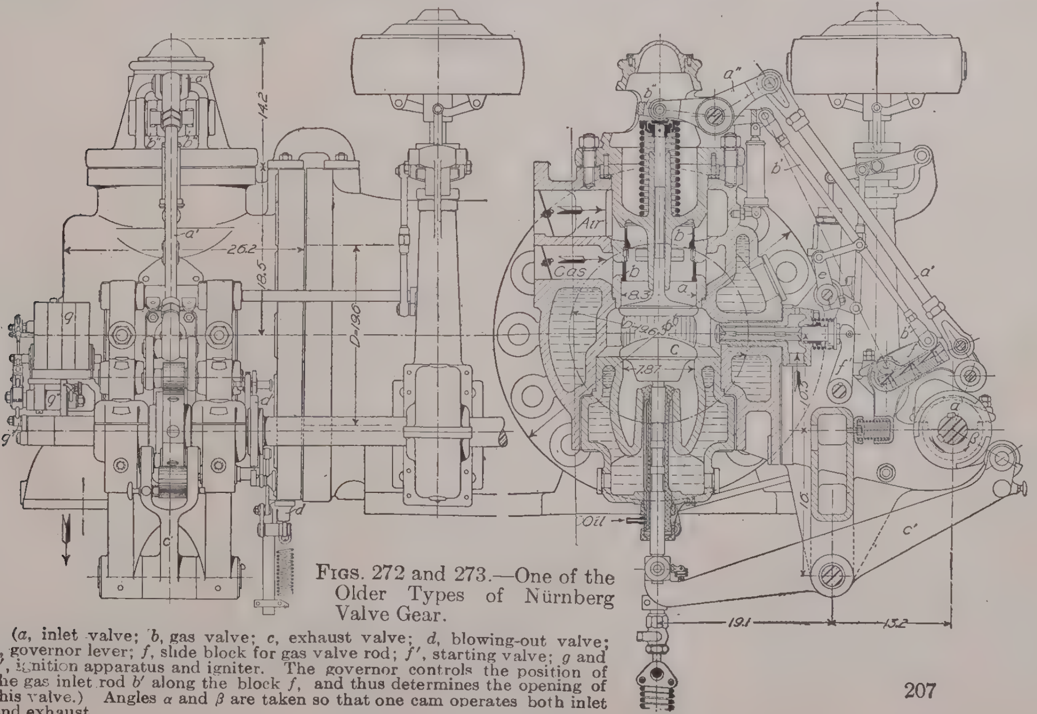
Fig. 269.—Valve Gear, Old Type, Gasmotorenfabrik, Deutz.

(*a*, inlet; *b*, gas; *c*, exhaust valve; *d*, open hot tube igniter. The valve levers are lettered the same as their respective valves.)



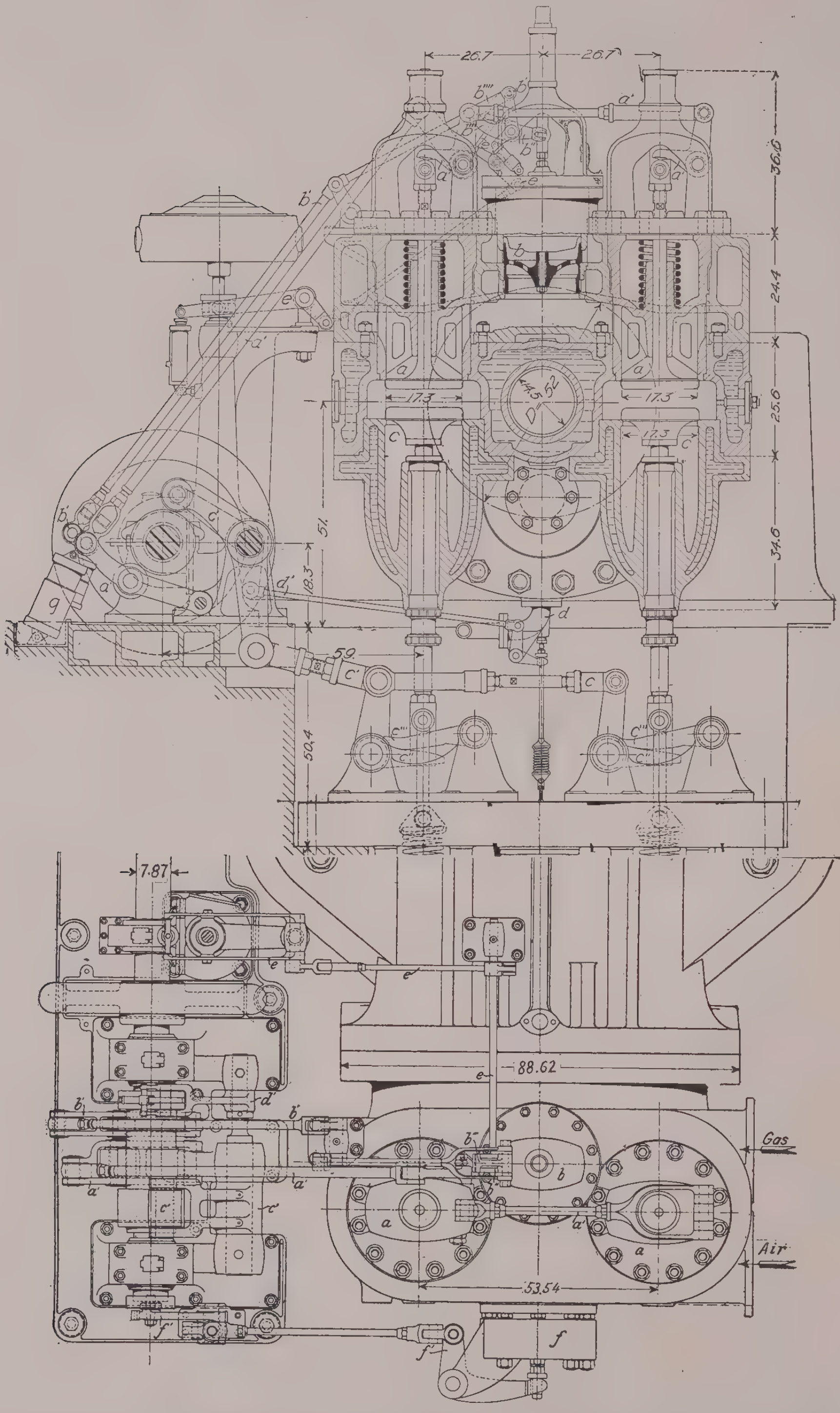
Figs. 270 and 271.—Valve Gear, Motorenfabrik Werdau.

(*a*, inlet; *b*, gas; *c*, exhaust valve; *d*, electric igniter; *d'*, make-and-break cam; *e*, igniter; *f*, lever for regulation of air supply.)



Figs. 272 and 273.—One of the Older Types of Nürnberg Valve Gear.

(*a*, inlet valve; *b*, gas valve; *c*, exhaust valve; *d*, blowing-out valve; *e*, governor lever; *f*, slide block for gas valve rod; *f'*, starting valve; *g* and *g'*, ignition apparatus and igniter. The governor controls the position of the gas inlet rod *b'* along the block *f*, and thus determines the opening of this valve.) Angles α and β are taken so that one cam operates both inlet and exhaust.



FIGS. 274 and 275.—Valve Gear used on some of the older Type of 1500–2000 H.P. Nürnberg Tandem Engines.

(*a*, inlet valves; *b*, gas valve; *c*, exhaust valves; *d*, blowing-out valve; *e*, governor linkage; *f*, starting valve; *g*, valve lever springs. These engines are governed by adjusting the port openings through gas valve *b*, the governor through *e* changing the fulcrum about which lever *b″* turns, and thus controlling the valve lift.

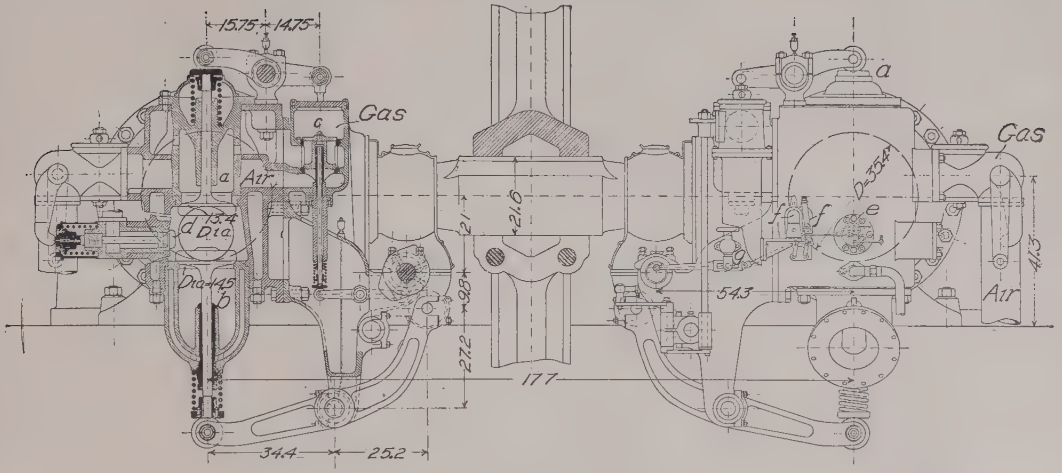


FIG. 276.—Valve Gear of a 600 H.P. Twin Engine, L. Soest & Co., Düsseldorf-Reisholz. (*a*, inlet valves; *b*, exhaust valve; *c*, gas governing valve; *d*, starting valve; *e* and *f*, double electric ignition; *g*, spark control. See Plate XXIII, Part III.)

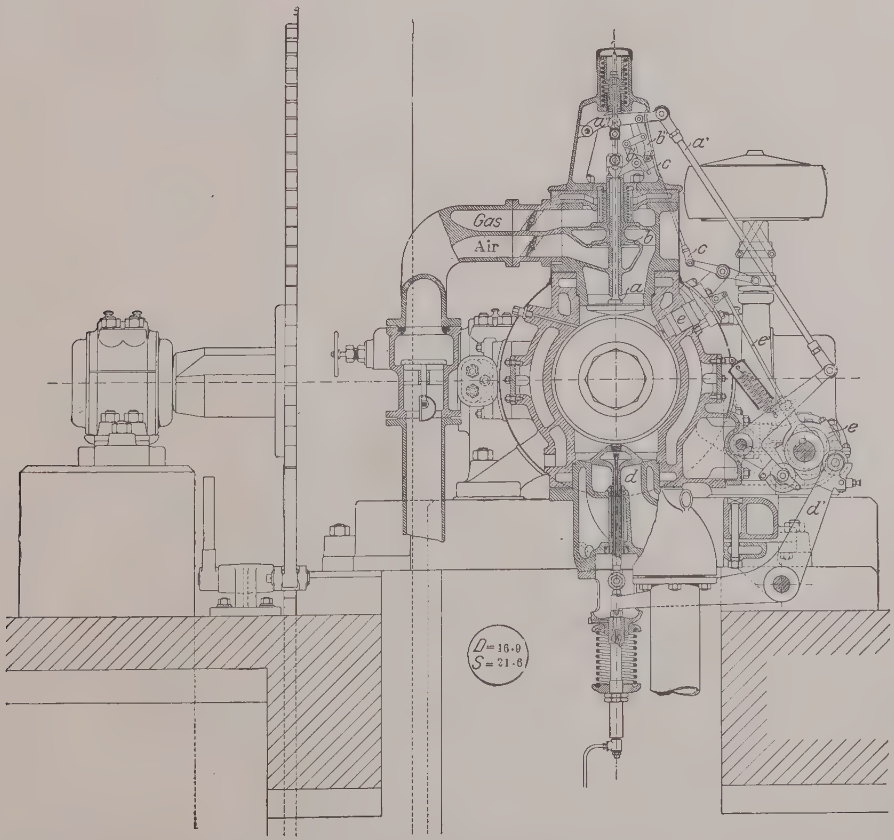
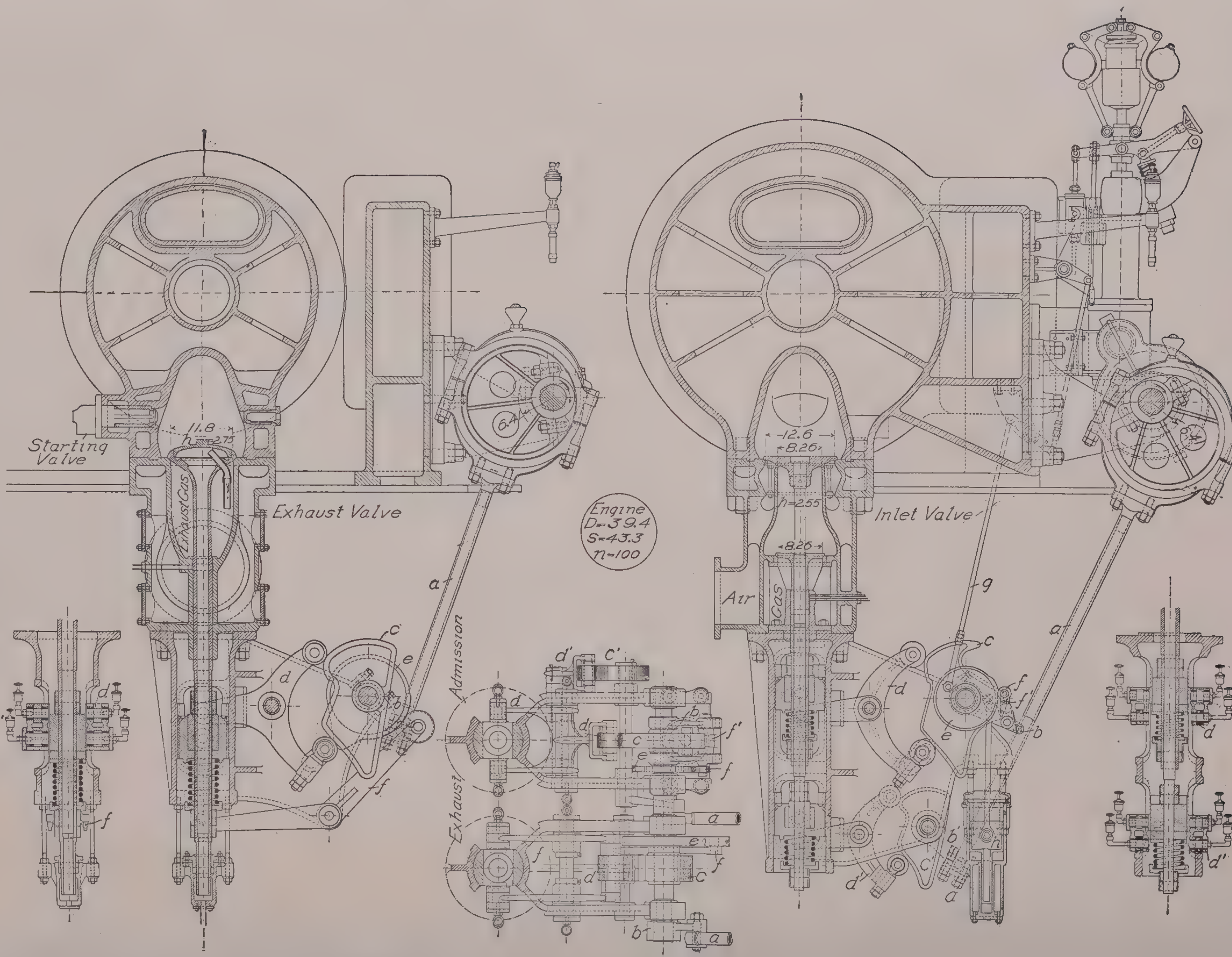


FIG. 277.—Valve Gear for a 100 H.P. Double-acting 4-cycle Engine, Langen & Wolf, Vienna. (*a*, inlet; *b*, gas; *d*, exhaust valve; *e*, igniter. The corresponding operating levers are marked with the same letters primed. Lever *b*′, fastened to *a*′, is thrown to the left by guide *c*, lifting the gas valve *b*. The governor, through rod *c* changes the position of the guide block *c*, thus controlling the left of *b*.)

FIGS. 278 – 282. — Valve Gear, Simplex-Cockerill Engine.

At the left, exhaust gear: Eccentric rod *a*, fastened to the fixed lever *b*, positively opens and closes the exhaust valve through the oscillating cam *c* and the bell-crank lever *d*. Near the end of the stroke cam *e*, through the lever *f*, puts the small auxiliary spring in compression.

At the right, inlet gear: Eccentric rod *a* is connected to both levers *b* and *b′*. The former is loose on its shaft, the latter is fastened. *a* operates the gas inlet valve through cam *c* and bell-crank *d*. Cam *c* oscillates with its shaft or is released from it, depending upon the action of the latch *f*, which connects *b* with *c*. The latch in turn is controlled by the action of the knock-off cam *e*, whose position is determined by the governor through *g* depending upon the load on the engine. *h* is an air dash-pot controlling the seating of the gas valve gear when the latch releases it. The main inlet valve is operated through *b′*, *c′* and *d′* in a manner similar to that used for the exhaust valve. See further illustrations in Part III.



Constructive Details.

TABLE 19

DIMENSIONS OF TEETH OF SCREW GEARS

Diametral Pitch.	Circular Pitch.	Depth of Tooth.		Lateral Pitch $t_s=\frac{t_n}{\cos\alpha}$									
				$\alpha=26° 35'$		$\alpha=30°$		$\alpha=45°$		$\alpha=60°$		$\alpha=63° 25'$	
$=\frac{t_n}{\pi}=M$	$=t_n$ Inches	Total Inches	Face Inches	t_s Inches	$\frac{t_s}{\pi}$ Inches	t_s Inches	$\frac{t_s}{\pi}$ Inches	t_s Inches	$\frac{t_s}{\pi}$ Inches	t_s Inches	$\frac{t_s}{\pi}$ Inches	t_s Inches	$\frac{t_s}{\pi}$ Inches
.08	.251	.173	.08	.280	.089	.289	.091	.356	.113	.502	.16	.560	.178
.10	.314	.216	.10	.351	.112	.360	.114	.444	.141	.628	.20	.700	.223
.12	.377	.260	.12	.421	.134	.434	.138	.533	.170	.754	.24	.841	.268
.16	.503	.347	.16	.563	.179	.580	.184	.713	.227	1.006	.32	1.124	.358
.20	.628	.432	.20	.702	.224	.725	.231	.890	.283	1.256	.40	1.404	.447
.24	.754	.521	.24	.842	.268	.869	.277	1.066	.339	1.508	.48	1.684	.537
.28	.880	.607	.28	.983	.313	1.016	.323	1.244	.396	1.760	.56	1.966	.625
.32	1.006	.693	.32	1.125	.358	1.219	.388	1.422	.453	2.012	.64	2.249	.717
.36	1.131	.780	.36	1.263	.402	1.305	.415	1.599	.508	2.262	.72	2.528	.806
.40	1.256	.864	.40	1.405	.447	1.449	.461	1.774	.562	2.512	.80	2.807	.892
.44	1.362	.953	.44	1.544	.491	1.595	.507	1.954	.622	2.764	.88	3.088	.982
.48	1.508	1.039	.48	1.685	.537	1.741	.554	2.132	.678	3.016	.96	3.371	1.072
.52	1.635	1.127	.52	1.823	.580	1.885	.599	2.308	.734	3.266	1.04	3.650	1.160
.56	1.759	1.213	.56	1.964	.625	2.030	.646	2.485	.791	3.518	1.12	3.932	1.250
.60	1.885	1.296	.60	2.108	.671	2.176	.692	2.665	.847	3.770	1.20	4.215	1.340
.65	2.042	1.408	.65	2.280	.726	2.358	.750	2.886	.918	4.084	1.30	4.565	1.452
.70	2.198	1.516	.70	2.456	.781	2.537	.806	3.107	.988	4.396	1.40	4.913	1.563
.75	2.356	1.625	.75	2.634	.838	2.719	.865	3.330	1.058	4.712	1.50	5.266	1.672
.80	2.512	1.732	.80	2.808	.893	2.900	.923	3.551	1.128	5.024	1.60	5.616	1.787
.85	2.670	1.842	.85	2.984	.948	3.083	.981	3.774	1.200	5.340	1.70	5.969	1.896
.90	2.826	1.949	.90	3.158	1.006	3.263	1.038	3.995	1.271	5.652	1.80	6.317	2.017
.95	2.984	2.057	.95	3.333	1.060	3.445	1.097	4.218	1.340	5.968	1.90	6.670	2.123
1.00	3.142	2.167	1.00	3.511	1.117	3.628	1.155	4.445	1.414	6.284	2.00	7.024	2.236

1. Driving Gears. The lay shaft is usually operated by means of cylindrical screw gears, of which the driver, on the crank-shaft, is generally steel, the driven gear may be bronze or cast iron. In case the width of the teeth is ample and oil-bath lubrication is employed, both gears are sometimes milled out of cast iron. On account of the comparatively small surface of contact and the sliding friction always present in this type of gear, constant lubrication is practically a necessity. In order to simplify the construction of the bearings, it is desirable to have the same diameter of pitch circle for both gears, or at least to make the diameter of the driven gear as small as possible, in spite of the fact that the speed ratio is 2:1. For that reason a common value for the angle α (Fig. 283) is 63° 25′, or 60° in round numbers, for the driving gear, and $\alpha=26° 35'$, or approximately 30°, for the driven gear. The lateral pitch t_s, if t_n is the normal pitch (see Fig. 283), will then be

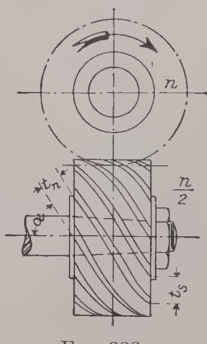


FIG. 283.

$$t_s=\frac{D_t\pi}{z}=\frac{t_n}{\cos\alpha}. \qquad (1)$$

This for the driving gear becomes, when

$$\alpha = 63^\circ\ 25', \quad \text{or} \quad \alpha = 60^\circ,$$

$$t_s = \frac{t_n}{.4473} = 2.236\ t_n, \qquad t_s = \frac{t_n}{.50} = 2.0\ t_n,$$

and for the driven gear we will have, when

$$\alpha = 26^\circ\ 35', \quad \text{or} \quad \alpha = 30^\circ,$$

$$t_s = \frac{t_n}{.8946} = 1.118\ t_n, \qquad t_s = \frac{t_n}{.8660} = 1.155\ t_n.$$

The pressure on the teeth of the driven gear causes a thrust on the lay shaft. This is less with the smaller value of α for this gear.

Table 19, page 211, gives some of the main proportions of screw gears.

The width of the teeth is from 2–$2.5t_n$, the thickness at the pitch circle about $.5t_n$, that is, there is no back-lash. For practical values of t_n see Table 20. Putting the diametral pitch $\frac{t_n}{\pi} = M$, we may also write

Total depth of tooth $= 2\frac{1}{6}M$, depth above pitch line, i.e., face of tooth, $h' = M$;

Pitch diameter $D_t = \frac{Zt_s}{\pi}$, where $Z =$ number of teeth;

External diameter of gear, $D_a = D_t + 2\frac{t_n}{\pi} = D_t + 2M$.

TABLE 20

AVERAGE DIMENSIONS OF VALVE-GEAR PARTS

Normal Pitch of Screw Gears $= t_n$, Dia. of Lay Shaft $= d_s$, Dia. of Cam Rollers $= d_r$.

Nominal B.H.P. of Engine.	2	5	10	15	20	30	40	50	60	75	100
t_n inches	.648	.680	.680	.711	.711	.742	.742	.804	.804	.866	.926
d_s "	1.25	1.37	1.37	1.62	1.62	1.75	1.75	2.00	2.00	2.25	2.50
d_r "	1.62	1.75	2.00	2.25	2.37	2.62	2.75	3.00	3.25	3.50	4.00

2. Lay Shaft. The computation based upon the external forces acting upon the cams gives dimensions much too small for both lay shaft and transmission gears. Table 20 gives some practical values for these dimensions for engines up to 100 B.H.P. If the lay shaft is also called upon to actuate devices out of the ordinary, as, for instance, circulating pumps for cooling water or air compressors, the values for t_n and d_s in the table should be correspondingly increased, or, instead of screw gears, milled helicoidal gears should be used. These possess the advantage over screw gears in that the pressure is distributed over the surface of the flank of the tooth instead of bearing in one line only. In general, however, the lay shaft should not be called upon to do such work, as it reacts unfavorably upon the governor, especially if the power required is of fluctuating quantity. Even in ordinary operation it is often noticed in large machines that the varying torque to which the lay shaft is subjected during one revolution causes vibrations in the shaft itself, knocking in the gears and irregular governor action. The resisting torque in the lay shaft is a maximum at the time of opening of the exhaust valve. As the cam roller leaves the exhaust valve cam, the lay shaft receives an impulse through the action of the valve spring upon the valve lever. This causes a taking up of any back-lash in the driving gear if the torque so produced exceeds the frictional resistance of the shaft, which in large engines is nearly always the case. Several German designers therefore use a special fly-wheel on the cam shaft, which, by means of its inertia, prevents the reversal of the torque on the shaft, and hence contributes much to noiselessness of operation and steadiness of regulation. English and American designers very often eliminate the bad effect upon regulation of the varying torque in the lay shaft by the use of centrifugal governors in the fly-wheel, or by giving the governor an independent drive from the main shaft.

3. Cams. Fig. 284 shows a complete cam system consisting of three cams, *E* for the inlet, *A* for the exhaust, and *K* to reduce compression at starting. Since the lay shaft, sometimes called the half-time shaft, has only one half the angular velocity of the main shaft, the values of the angles given in Fig. 284 must be doubled in order to refer to crank angles. The lines of rise and fall of the cams should be tangential to the base circle, and the transition from the tangents to the outer surface of the cam should be as gradual as the valve lift curve (curve *h*, Fig. 255, p. 201) permits. It is possible, by properly fixing the position of the cam-rollers, to actuate both the inlet and exhaust valve from the same cam (see Fig. 273). This simplifies the gear a little, but it is questionable whether the gain realized is sufficient to balance the attendant inaccuracies in the motion of the valves. Regarding the direction of motion of the cam shaft, it will generally be found by inspection that the lift of the valve levers is more easy in one direction than in the other. This of course guarantees less wear and noise.

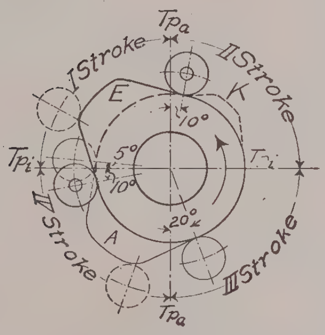
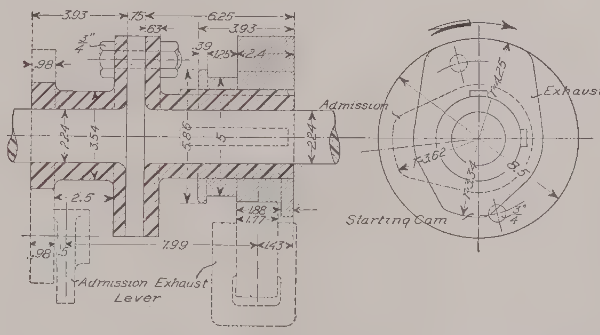


FIG. 284.—Normal or Standard Cam System.



FIGS. 285 and 286.—Cam System for a 40–50 H.P. Hornsby-Akroyd Engine.

Large cam diameters permit of accurate adjustment of the valve motion, but knocking against the cam rollers is likely to result when the peripheral velocity of the cam approaches 3 ft. per second. Large cam diameters also increase the torque in the shaft. The method of fastening the cam to the shaft requires careful consideration. Cams movable in an axial direction have not proven reliable in large engines unless the method of fastening is very carefully designed. An example of this is given in Figs. 285 and 286.

The cam rollers are made of hardened steel. Common dimensions are given in Table 20. The width of the roller for the inlet cam may be made $=.3\,d_r$, that for the exhaust cam $=.4\,d_r$. These dimensions, however, should be checked to see that the maximum pressure does not exceed 3000 lbs. per inch of width of roller.

4. Valve Levers. In order to keep the inertia forces as low as possible, the cross-section of the valve levers should be such as to combine greatest strength and stiffness, especially in the horizontal plane, with minimum weight. The return motion of the levers should be produced by special springs, and should not be left to the action of the valve springs. Suppose that the weight of the actuated parts of the valve gear is G lbs., referred to the center of gravity, and let the distance the latter is to be moved in t seconds be s, then the tension of the special return spring, also referred to the center of gravity as the point of application, should be at least

$$P_m=\frac{M\times 2s}{t^2}\mu=\frac{G\times 2s}{32.2t^2}\mu\sim\frac{Gs\mu}{16t^2}\text{ lbs.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

In this equation μ is a correction factor which takes into account frictional resistances and unavoidable errors of computation. Practice has shown that it may be taken at $\mu=1.25$ to 1.50.

The computations for strength of the valve levers, which hardly require any further explanation, should be based on a pressure at the moment of opening of at least 30 lbs. per sq.in. for the inlet and 75 lbs. per sq.in. for the exhaust valve. It is possible, in engines using compressed air for starting, that through careless handling of the starting valve (taking air full stroke) the opening resistance of the valves may be in excess of the above figures.

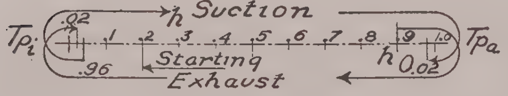


FIG. 287.—Normal or Standard Valve Setting Diagram.

5. Adjustment of the Valve Gear. The valve gear may be quickly set with the aid of diagrams like Fig. 284 or 287. In either of these T_{pi} and T_{pa} designate respectively the inner and outer dead center position of the piston. To facilitate the operation of adjusting the valve gear in the shop it is well to fasten definitely only the gear on the main shaft and the cams, but to leave the gear on the lay shaft free until the final adjustment is made. To this end the seat of this gear may be turned

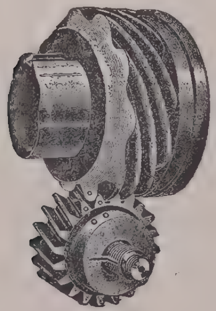


FIG. 288.

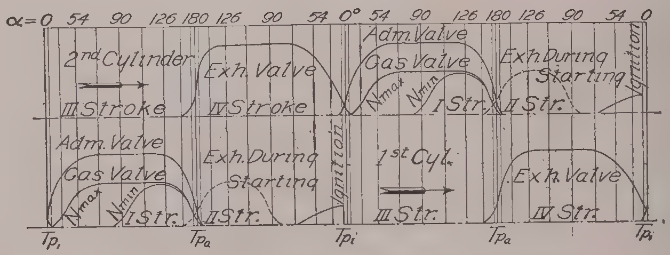


FIG. 289.—Valve Lift Diagram for a Twin Engine.

slightly conical, in the ratio, say of 1:12 or 1:10, and during the adjustment the gear is kept from slipping by forcing it upon the cone by means of a jam nut. After the setting is completed and the engine finally tested out, the pair of gears should be marked as indicated in Fig. 288, and the driven gear may then be finally keyed in place.

In designing the valve gears for multi-cylinder engines, a clear insight into the sequence of the various valve actions in all of the cylinders may be gained by drawing the valve lift curves in their proper phase with reference to one system of coördinates. In Fig. 289 this has been done for a two-cylinder machine.

IX. Fly-wheels

Material. In stationary engines usually cast iron throughout; in the largest sizes of engines sometimes cast iron with steel or cast-steel arms; for high-speed automobile engines now and then steel or cast-iron body with lead rim.

Allowable Bending Stress K_b, on account of uncertainty of the nature of the stresses encountered and to guard against unforeseen contingencies, not to exceed 3000 lbs. per sq.in. for cast iron or 7000 lbs. per sq.in. for soft steel.

1. Determination of Weight of Fly-wheel. In theoretical investigations it is usual to employ the tangential effort diagram obtained from an assumed or real indicator diagram, to compute the moment GD^2_s or the moment of inertia of the rim.

$$I=\frac{GD^2_s}{4g}\sim\frac{GD^2_s}{128}=\frac{GR^2}{g}, \qquad \ldots\ldots\ldots\ldots \quad (1)$$

which is required for any given coefficient of regulation δ. This method, however, is quite complicated and takes time. For that reason it is not much used in the drafting room, but since it gives a clear insight into the pressure and velocity changes it will be explained in detail. This will be followed by the development of a simple and purely mathematical method of equal practical utility.

Plates II, III, and IV give a number of accurate tangential effort diagrams for single-cylinder engines; Plates V and VI similar diagrams for the common multi-cylinder combinations. All of these were constructed from actual indicator diagrams,[1] for the weight G_t or G_0 of the reciprocating parts, noted on the plates, and for connecting-rod lengths equal to $L=5\,r$. All diagrams are drawn for the full operation, that is, for two revolutions for the 4-cycle, and for one revolution for the 2-cycle. In the case of multi-cylinder engines it is usually sufficient to draw the diagram for the interval between two successive ignitions. To construct these diagrams the first operation is to redraw the lines of the indicator diagram, placing the various events side by side in a row, and spacing off the ordinates above or below the zero pressure line as required. Next draw the diagrams for the inertia forces according to page 156, and lay them in over the lines of the developed indicator diagram in proper order. The combination of the indicated pressures with the inertia forces then gives the diagram of net piston pressures. These in turn are used in the determination of the tangential effort diagram, which has for its length of base the developed crank circles of a complete cycle. The abscissæ for the ordinates on all the diagrams are equivalent to 10° crank angle, and corresponding piston and crank positions are marked by the same figures.

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[1] Published in the Zeitschrift d. V. D. I., 1901, p. 369.

To determine the ordinates of the tangential effort diagram any of the well known graphical methods may be employed. Mathematically, their height is

$$p_t = p\frac{\sin(\alpha+\beta)}{\cos\beta} \text{ lbs. per sq.in.}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

in which p is the net piston pressure at any given instant, α = the crank angle, and β the connecting-rod angle.

To obtain the resistance diagram, we may assume that the net effective effort resulting from the fluctuating effort of one complete cycle is used up by a uniform resistance W acting tangentially at the crank circle. This diagram therefore is a rectangle whose base b is the length of the developed crank circles of one complete cycle, and whose height p_w may be determined from the mean effective pressure p_i corresponding to the nominal indicated horse-power, to which the work of resistance must evidently be equal.

The net effective work A_i is the difference between the total indicated work A_a and the work of compression A_c, and is produced during one stroke of the piston. Hence

$$A_i = A_a - A_c = p_iF \times 2r \text{ ft.-lbs.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

The uniform resistance encountered is

$$W = p_wFb \text{ ft.-lbs.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

The length b depends upon the type of engine as follows:

For single-acting 4-cycle engines, $b = 4\pi r$;
For single-acting 2-cycle engines, $b = 2\pi r$;
For double-acting 2-cycle engines, $b = \pi r$.

Since

$$A_i = W \quad \text{or} \quad p_iF \times 2r = p_wFb, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3a)$$

we obtain, by substituting the proper values of b, the following values of p_w for all single-cylinder engines.

4-cycle engines, $p_w = \frac{p_i}{2\pi}$; . . . . (5)

Single-acting 2-cycle engines, $p_w = \frac{p_i}{\pi}$; . . . . (5a)

Double-acting 2-cycle engines, $p_w = \frac{2p_i}{\pi}$. . . . . (5b)

The crank travel b between the successive ignitions is therefore determined in the various cases by the following crank angles: in the double-acting 2-cycle, 180°; the single-acting 2-cycle, 360°, and the single-acting 4-cycle, 720°. In what follows the pressures p_i, p_c, and p_w will be expressed in pounds per square inch. If F is the effective piston area in square inches, the factors p_iF, p_cF, and p_wF represent the value of the total piston or resisting pressures P_i, P_c, and P_w acting at any given instant.

The difference existing at any given instant between the variable tangential effort and the uniform resistance is shown in the diagrams by the relative positions of the two lines representing these forces. Where the tangential effort curve lies above the line of uniform resistance, we have an excess of energy furnished; similarly, where the curve cuts below the line of resistance, a deficiency exists. As is well known, this deficiency is made up by means of the kinetic energy K derived from the energy excess A and stored in the revolving mass M of the fly-wheel. The energy exchange between crank and fly-wheel takes place under constantly varying angular velocity of the crank. The velocity diagram (Fig. 4, Plate II) gives some idea of the kind and magnitude of the velocity variations. It will be noted that the maximum velocity $V_{\max}$ of the crank pin is found at that point of the diagram (Fig. 3, Plate II) where the tangential effort cuts below the line of uniform resistance, that is, near the end of the expansion stroke. Beyond this stroke there is in general a steady drop in the velocity, interrupted only here and there by the effect of inertia forces of the reciprocating parts, until the minimum velocity, $V_{\min}$ of the pin is reached just after compression. After this point the tangential effort curve quickly rises again above the line of uniform resistance owing to the combined action of the inertia forces and the new combustion.

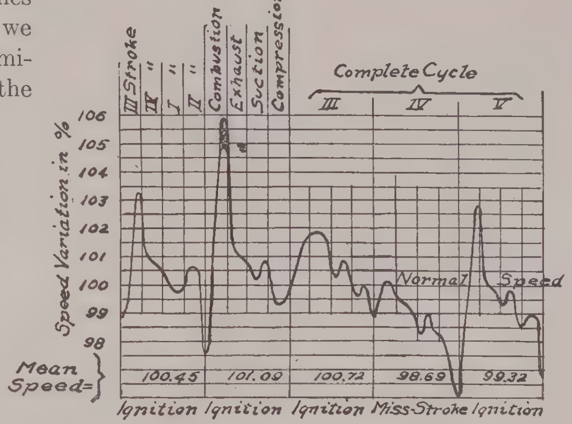
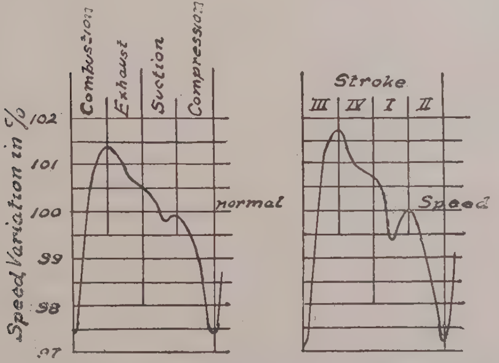


Fig. 290.—Tachogram, 100 H.P. Crossley Engine.



Figs. 291 and 292.—Tachograms, 20 H.P. Diesel Engine.

Actual tachometer records, taken on a large scale with very sensitive tachometers, corroborate throughout the statements above made in regard to velocity variation of crank pin as derived from the constructed tangential effort diagram. Fig. 290 shows a record for five complete cycles as obtained from a 100 H.P. single cylinder Crossley-Otto producer gas engine. This engine had two fly-wheels, each weighing 5 tons, and was supposed to make normally 180 turns per minute. During the tachometer tests the average r.p.m. was 164 at a load of about 60 H.P. The velocity variations shown in this diagram are remarkable for irregularity, magnitude, and unexpected occurrences. This is true not only as regards variation between cycles, but also within the individual cycles. It is evident that the fly-wheels were too light at the lower speed used and that the individual explosions were of very unequal strength. Much more regular, that is, uniform, are two records from a 20-H.P. Diesel engine, running normally at 180 r.p.m., shown in Figs. 291 and 292. The fly-wheel in this engine weighed about 3700 lbs. Fig. 291 is taken at 9.36 B.H.P. and 292 at 21.75 B.H.P.[1]

Assume that

$$A=K=M\frac{v^2_{\max}-v^2_{\min}}{2}\text{ ft.-lbs.} \quad \quad (6)$$

If we let $v=\frac{v_{\max}+v_{\min}}{2}$ = the mean velocity of the crank pin and the coefficient of regulation

$$\delta=\frac{v_{\max}-v_{\min}}{v}=\frac{n_{\max}-n_{\min}}{n}, \quad \quad (7)$$

equation (6) may be written

$$A=K=Mv^2\delta. \quad \quad (6a)$$

Further, if the velocity V in ft. per second of the center of gravity of the cross-section of the wheel rim be substituted for v, and the weight G of the rim be put in place of M, we finally have

$$A=K=\frac{G}{g}V^2\delta. \quad \quad (8)$$

Assume that the area representing energy excess or deficiency in the tangential effort diagram contains f sq.in. The energy value of this area will be

$$A=fFm\frac{S}{l}\text{ ft.-lbs.} \quad \quad (8a)$$

In the above equation $S=2r$ = stroke in feet, l = scale length in inches of one stroke in the net piston pressure diagram, and m = number of pounds pressure per inch of ordinate of this diagram.

The necessary weight G of the fly-wheel rim must be, from eq. (8),

$$G=\frac{Ag}{\delta V^2}=\frac{Ag}{\delta\left(\frac{\pi Rn}{30}\right)^2}\sim\frac{2900\,A}{\delta(Rn)^2}\text{ lbs.} \quad \quad (9)$$

From (9) we also derive

$$\delta=\frac{Ag}{GV^2}=\frac{2900\,A}{G(Rn)^2}. \quad \quad (10)$$

Under ordinary conditions the area of excess in the tangential effort diagram is equal to the area of deficiency, in which case the latter may be left out of consideration. It may happen, however, in engines governing on the hit-and-miss principle that that part of the area of the tangential effort diagram lying below the line of resistance exceeds the positive area of excess. In such a case the value of A must be determined from the deficiency area. This will be explained more in detail when we come to consider various methods of regulation. In Plates V and VI, showing diagrams for 2-cycle and multi-cylinder engines, the areas upon which the computation of A should be based have in each case been indicated by marginal hatching.

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[1] These diagrams, and also those on page 227 and 228, were taken from an article by Prof. A. Reeve in the *Engineering News* of Aug. 2, 1900. They were made by the use of an electrically excited tuning-fork, the stylus of which passed over smoked paper. The records so obtained were afterwards carefully copied.

At the outset it may be stated that for **4-cycle engines** the inertia effects of the reciprocating parts may be neglected in fly-wheel computations. It is true, of course, that these inertia pressures tend to decrease the pressures on the crank pin at the moment of maximum compression and combustion pressures; it is also true that the inertia effects cause the tangential effort to be more uniform during the second and third strokes of the cycle; but all of these influences affect the coefficient of regulation δ of the engine but very little. The influence that the moving parts may have upon the area used as a basis of computation for the value of G or δ is very insignificant. The difference usually amounts to but from .5 to .7%, and only in the most unfavorable cases does it amount to 1% of the true area.[1] The explanation of this not quite self-evident fact is found in the circumstance that the line of uniform resistance, which forms the lower boundary of the excess energy area, lies very close to the line of zero pressure in 4-cycle engines. In fact, this line cuts the tangential effort curve of the expansion stroke close to the dead centers. In these piston positions, however, the crank angle is so small that the inertia forces do not produce any considerable turning effort, and the pure tangential pressure curve is therefore influenced but little. Since the positive inertia pressures during any one stroke must equal the negative pressures, i.e., their sum$=0$, the absolute size of the work area is not affected by them. Their only effect is to change the areas above and below the line of uniform resistance by changing the shape of the circumscribing curves. But the amount of this change is the smaller the closer the line of resistance is to the zero line. In the case of 4-cycle engines the distance p_w from the zero line to the uniform resistance line is so small that the designer is justified in neglecting the effect of the inertia forces in fly-wheel computations for engines of this type. (Of course the value of the correct tangential effort diagram in judging actual pressure conditions and reversal of pressure is not in any way affected by these considerations.)

The mean pressure of the negative work done during the first and fourth strokes in drawing in the fresh charge and discharging the exhaust gases is only about 1% of the mean pressure of the absolute positive work in modern large engines with mechanically operated valves of large cross-section. This work has therefore practically no effect upon the turning moments. In any case its effect upon the coefficient of regulation is less than that of the frictional resistance of piston and cross-head. But so little is definitely known about the magnitude and distribution of the latter over the four strokes of the cycle, that its effects cannot be taken into account with any accuracy in fly-wheel computation. For that reason we are justified in this investigation in also neglecting the negative work of the suction and exhaust strokes, and confining our attention merely to the negative work A_c of the compression stroke and the absolute positive work A_a of the combustion and expansion stroke.

The difference $A_a - A_c$ represents the indicated effective work A_i, which, according to eq. (3*a*), must be equal to the uniform work of resistance W. In the tangential effort diagram of a 4-cycle machine (Fig. 293), A_a is represented by the positive area *abcde*. After subtracting the area of the strip $A_0=abde$, the remainder is the area $A=bcd$, from which is determined the rim weight G of the fly-wheel. The area A_0 is smaller than the parallelogram aa_1e_1e only by the small corner areas $x+y=aa_1b+de_1e$. The area aa_1e_1e represents one quarter of the work of the uniform resistance $\left(=\frac{W}{4}\right)$. The integration of a large number of turning effort diagrams of 4-cycle engines has conclusively shown that the area of the two small triangles x and y, cut from the parallelogram aa_1e_1e rarely exceeds 1%, and in the extreme does not amount to 1½% of the positive work area *abcde*. The influence of these areas upon the result of fly-wheel computations is therefore in all cases very small.[1] *It is permissible, therefore, without introducing undue errors, to determine the excess energy A directly from the expression* $A_a-\frac{W}{4}$. *Now both* A_a *and* $\frac{W}{4}$ *can be taken directly from an indicator diagram, and the latter is consequently quite sufficient to determine the amount of excess energy A and finally also the weight G of the fly-wheel rim.* This enables us to substitute a simple mathematical method for the graphical method.

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[1] Actual proof of this may be found in the Zeitschrift d. V. D. I., 1901, p. 371.

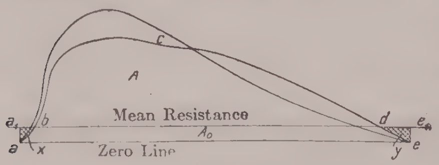

FIG. 293.

The term "absolute positive work" (A_a) is not much used in actual practice. The usual method is to planimeter the indicator card and to determine from this the value of A_i, that is, the net indicated work, and for that reason it seems best in the method to be developed to use the more common terms A_i (in ft.-lbs.) and N_i (in H.P.).

Let the mean pressure of the compression stroke be p_c, and the mean effective pressure (of the net indicated work) be p_i lbs. per sq.in. The absolute positive work will then be

$$A_a=A_i\left(1+\frac{p_c}{p_i}\right)\text{ ft.-lbs.} \quad \quad (11)$$

Now, putting

$$\rho=\frac{p_c}{p_i}, \quad \quad (12)$$

we may write

Excess energy $\quad A=A_a-\frac{W}{4}=A_i(1+\rho)-\frac{W}{4}\text{ft.-lbs.}$

But from eq. (3*a*) $\quad A_i=W=p_wFb.$

Hence

$$\begin{aligned} A&=W(1+\rho)-\frac{W}{4}=(.75+\rho)W \\ &=(.75+\rho)p_wFb \\ &=(.75+\rho)p_wF\,4\,r\,\pi\text{ ft.-lbs.} \quad \quad (13) \end{aligned}$$

for a 4-cycle machine.

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[1] For numerical proof, see Zeitschrift d. V. D. I., 1901, p. 371.

The uniform turning moment $p_wFr=P_wr$, however, may be expressed by the general dynamic equation,

$$Pr=\frac{550N}{\omega}=\frac{60\times550N}{2\pi n}=5255\frac{N}{n}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (14)$$

Hence eq. (13), in connection with eq. (8), becomes

$$(.75+\rho)\,5255\frac{N_i}{n}4\pi=A=\frac{G}{g}V^2\delta,$$

or

$$G=\frac{(.75+\rho)5255\,N_i4\pi g}{V^2\delta n}=\frac{2124600(.75+\rho)\,N_i}{\delta V^2n}\text{ lbs.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (15)$$

In round numbers we may therefore write

$$G=\frac{2125000\,N_i(.75+\rho)}{\delta V^2n}\text{ lbs.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (16)$$

and

$$\delta=\frac{2125000\,N_i(.75+\rho)}{GV^2n}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (17)$$

It is easier for both designer and constructor to use the radius R or the diameter D_s of the gravity circle of the wheel rim instead of the velocity V in computation. Substituting therefore

$$V=\frac{\pi D_sn}{60}=\frac{D_sn}{19.10}$$

in eqs. (16) and (17), we finally have

$$G=\frac{77.5\times10^7(.75+\rho)\,N_i}{\delta D_s^2n^3}\text{ lbs.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (16a)$$

and

$$\delta=\frac{77.5\times10^7(.75+\rho)\,N_i}{GD_s^2n_3}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (17a)$$

Of the total weight G about $\frac{1}{10}$ is usually assigned to the arms of the wheel, so that only $\frac{9}{10}G$ needs to be considered in designing the rim. Investigations on gas engines, however, have shown that the influence of the weight of the arms seems to be greater than the above proportion would lead us to expect, that is, the experimentally determined coefficient of regulation is smaller than a rim weight of $\frac{9}{10}G$ should theoretically give. The reason for this may be in the fact that the arms are usually designed much heavier than is common in steam-engine construction.

It should be noted in eq. (17a) that the factor n enters in the third power, hence a comparatively small decrease in n increases the value of δ seriously. On the other hand, the same facts permit us to materially better the regulation of any given machine by a comparatively minor increase in the number of revolutions per minute.

The coefficient ρ, eq. (12), does not depend upon the constructive differences in various machines as much as it does upon the heating values of the fuel employed and the quality of the mixture. The purer and richer the mixture, the smaller will be the value of ρ, because the relation between work of compression and of expansion is under such conditions the most favorable. With the high compression pressures used to-day in the best machines, however, one is compelled to use comparatively lean mixtures to prevent pre-ignition and its attendant trouble. These mixtures burn with comparatively small development of work area, and upon this depends the fact that in general the value of ρ varies directly as the compression pressure, rising and falling with the latter.

To give some idea of the value of ρ, in approximate or preliminary computations for which ρ has not yet been definitely determined, the author investigated a considerable number of actual diagrams from the most important types of engines with a view to determining the values of A_i and A_c. Excluding the extreme cases, the following limiting values of ρ are found from the data so obtained:

TABLE 21

PRACTICAL VALUES OF $\rho=\frac{p_c}{p_i}$

Illuminating-gas engine	$\rho=.25$ to $.35$	Gasolene engines	$\rho=.10$ to $.20$
Producer-gas engine	$\rho=.35$ to $.45$	Alcohol engine	$\rho=.25$ to $.32$
Kerosene engine	$\rho=.30$ to $.40$	Diesel oil engine [1]	$\rho=.48$ to $.52$

The value of N_i, in the case of engines governing down to no load on systems *other than the hit-and-miss*, should be based upon the intended maximum horse-power, $N_{e\,\max}$, of the engine, which should be about 10% greater than the nominal horse-power N_n. If, as usual, the latter is determined at the brake, the mechanical efficiency η_m, which in the various constructed types varies from .70 to .85, must be taken into account. We may then write

$$N_i=\frac{N_{e\,\max}}{\eta_m},$$

in which $N_{e\,\max}$ stands for the maximum brake horse-power desired.

In engines using so-called "*precision*" *regulation*, the governing is effected by a change in the mean pressure of the combustion stroke. In the tangential effort diagram therefore the excess positive area decreases as the load decreases. The

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[1] For these engines, Paul Meyer (Zeitschrift d. V. D. Ingenieure, 1901, p. 413) proposed the following special equation:

$$G=\frac{894.5\times .01\,\Sigma(A)}{\delta R^2n^2}\text{ kg.}\quad\text{and}\quad \delta=\frac{894.5\times .01\,\Sigma(A)}{GR^2n^2},$$

in which $\Sigma(A)$ represents the energy in m.-kg. absorbed by the fly-wheel, between the limiting velocities $V_{\max}$ and $V_{\min}$. This energy, for the ordinary type of Diesel engine, is approximately represented by

$$\Sigma(A)=v(.77p_i+4.1)\text{ cm.-kg.},$$

in which v is the stroke volume in ccm.

[Note that in these equations the units are metric throughout.]

distance, p_w, however, which measures the uniform resistance distributed over the four strokes of the cycle, decreases at a much slower rate. Hence the relation of excess area to resistance area grows more favorable as the load on the engine decreases. Consequently, also, the coefficient of regulation δ, computed for $N_{e\,\max}$, constantly improves as the load falls.

The conditions are much less favorable when the *hit-and-miss system* of regulation is used. Every decrease in the load in such an engine means an increase in the value of δ, that is, less close regulation, a fact to be borne in mind when a certain value of δ is to be maintained also at partial loads. The reduced diagram shown in Fig. 294 serves to explain this. It is drawn for half load, i.e., theoretically there should be one miss between every two ignitions. The positive work A_a developed during one combustion, considered as a constant turning force, is now distributed over eight strokes, instead of four as for full load, and the crank travel b_0 from ignition to ignition is $2\times 4\,r\pi$. But since, as before, the indicated effective work A_i must be equal to the uniform work of resistance $W=W_0=p_{w0}Fb_0$, the value of p_{w0} must vary inversely as b_0, i.e., it must be one half of its full load value. The line of uniform resistance is therefore that much nearer to the zero line, and the excess area $A=abc$ of the full load is, at half load, increased to $A_0=a_1abcc_1$. We then have

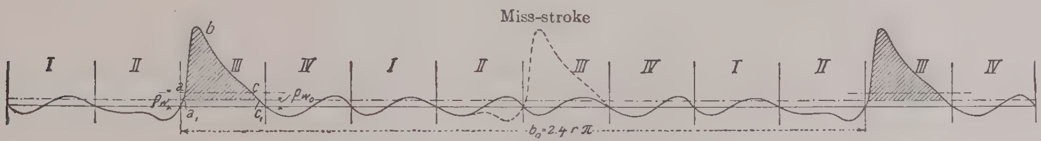

FIG. 294.

$$A_0=A_a-\frac{W_0}{2\cdot 4}, \qquad (18)$$

or, referring to eq. (13),

$$A_0=[(1-.125)+\rho]W_0=(.875+\rho)W_0. \qquad (18a)$$

The relation between the excess areas A and A_0 of the two load conditions, that is, in this case full load and half load, expresses directly the relation between the coefficients of regulation δ and δ_0. That is,

$${}_0=\frac{A_0}{A}\,\delta=\frac{.875+\rho}{.750+\rho}\delta. \qquad (19)$$

In the above discussion perfect governor action and ignition have been assumed; i.e., at half load, for example, one explosion must be regularly followed by one miss. If this is not the case, if, for instance, two, three, or more explosions are followed by a corresponding number of misses, the regulation will be still less close, since under these conditions the crank travel b_0 intervening between each governor action is further increased, the distance p_{w0} is further diminished, and excess area A_0 is still greater.

If an engine, at any partial load N_{i0}, works with x misses to one combustion, each of the latter must supply a crank travel equal to $1+x=m$ full four-stroke cycles or double revolutions. The coefficient of regulation then increases during the period of misses from the value δ at full load to

$$\delta_0=\frac{\left(1-\frac{.250}{m}\right)+\rho}{.750+\rho}\delta. \qquad (20)$$

If we assume an average value of $\rho=.30$, eq. (20) for various values of m, gives the following values of $\frac{\delta_0}{\delta}$.

$m =$	2	3	4	5	6	7	8	9	10
$\frac{\delta_0}{\delta} =$	1.120	1.158	1.178	1.190	1.197	1.203	1.207	1.210	1.213
	(1.120)	(1.150)	(1.166)	(1.177)	(1.184)	(1.190)	(1.194)	(1.197)	(1.200)

The values in parenthesis were obtained by measurement from a series of carefully constructed tangential effort diagrams, in which ρ and m correspond to the values assumed for eq. (20). The agreement in the two cases is evidently very satisfactory.

The field of application of eq. (20), however, is limited. It applies only in cases for which the tangential effort diagram shows a *positive* excess area equal to or greater than the *negative* excess area for *one* governor action.[1] If this is not the case, if, for instance, the negative areas, the hatched area in Fig. 295, exceed the positive excess area, then the former determine the weight of the fly-wheel, or, in the case of engines already constructed, the coefficient of regulation. Eq. (20) then no longer applies. The positive and negative excess areas for one governor action are almost exactly equal at $\frac{2}{3}$ load. In the full load diagram, Fig. 295, the positive area a is equal to

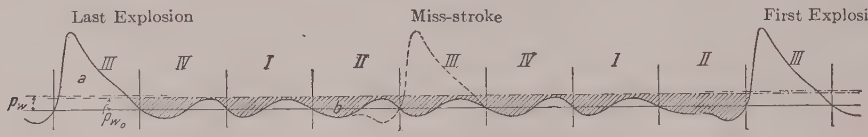

FIG. 295.

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[1] TRANSLATOR'S NOTE. In this discussion the terms "governor action" and "governing period" are used. In explanation, suppose that an engine of the hit-and-miss type operates under a given fractional load b, and that with perfect operation of the governor, which is here assumed, the engine regularly gives z explosions followed by y misses. Then $x+y=m$ is the "*governing period*," i.e., the number of double revolutions or complete cycles passed through while the fly-wheel goes from minimum to maximum and back to minimum velocity. It is evident that the load on the engine may then be expressed by $b=\frac{z}{m}$, and it is also evident that in one governing period the sum of the positive excess areas must always equal the sum of the negative excess, or deficiency, areas, otherwise the engine will either stop or run away.

By "*governor action*," on the other hand, is meant the period consisting of the last explosion stroke of a series plus the number of strokes concerned in the interval of miss cycles up to the beginning of the next explosion. It is evident that for one *governor action* the negative excess, or deficiency, areas may exceed the single positive excess area.

In the case of the partial loads represented by the series $\frac{1}{2}$, $\frac{1}{3}$, $\frac{1}{4}$, to $\frac{1}{m}$, where $z=1$ in all cases, the governing period is the same as the period of governor action.

the sum of the negative areas b. If now the load drops to $.9\,N_i$, for which—assuming perfect governor operation—there should be regularly 9 hits followed by 1 miss, the uniform resistance ordinate p_w will decrease to $\frac{9}{10}$ its full load value, and the positive excess area increases by $\frac{1}{10}\frac{W}{4}$. At the same time, however, the sum of the negative excess, or deficiency, areas also increases because these areas are now distributed over 7 instead of over only 3 strokes, as at full load. It follows that, as the negative areas increase at the expense of the positive excess area, during one governor action, the regulation coefficient increases, although immediately after the period of retardation it is brought down again to its best value as long as ignition regularly follows ignition.

Prof. R. Mollier,[1] investigating this question of hit-and-miss regulation along the lines above laid down, has developed the following special equation for the numerical determination of the variation of ∂ with different loads on the engine (see also eq. (18a)).

$$A_0 = A_i\left(1-\frac{1}{m}+\frac{3}{4}b+\frac{A_c}{A_i}\right). \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (\mathrm{I})$$

In this equation

A_0 = as before, the excess energy during the period of governor action;
$A_i = A_a - A_c$ = net work of expansion;
A_c = work of compression;
m = double revolutions (complete cycles) during one governing period;
$b = \frac{1}{m}$ = load on the engine.

Equation (I) thus assumes that each explosion is regularly followed by $a = m-1$ miss cycles. If, however, instead of one there should be z ignitions in one governing period, then $b = \frac{z}{m}$.[2]

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[1] See Zeitschrift d. V. D. I., 1903, p. 1704.

[2] TRANSLATOR'S NOTE. As an example, suppose an engine governs regularly | | | | | — —, i.e., five hits followed by two misses. Then $m=7$, $\frac{1}{m}=\frac{1}{7}$ and $b=\frac{5}{7}$. Hence, from eq. (I),

$$A_0 = A_i\left(1-\frac{1}{7}+\frac{3}{4}\cdot\frac{5}{7}+\frac{A_c}{A_i}\right) = A_i\left(\frac{39}{28}+\frac{A_c}{A_i}\right).$$

For full load, on the other hand, i.e., when no misses occur and $m=\infty$, eq. (13), page 220, gives

$$A = \left(.75+\frac{A_c}{A_i}\right)A_i$$

Putting $\frac{A_c}{A_i}=\frac{1}{4}$ in both equations, we have

for $\frac{5}{7}$ load $\qquad A_0 = \frac{23}{14}A_i,$

and for full load $\qquad A = A_i.$

Hence the excess area to be considered at $\frac{5}{7}$ load is greater than at full load, and if the fly-wheel is designed for full load, the coefficient of regulation ∂ at full load will increase at $\frac{5}{7}$ load to

$$\partial = \frac{A_0}{A}\partial = \frac{23}{14}\partial = 1.64\partial.$$

This is indicated in Fig. 296. It should be noted next that the condition that there shall be no misses at full load is ideal, and never realized in practice in hit-and-miss engines. A miss cycle

It is evident that the greatest safety against any change in δ exists when in eq. (I) the value of m is made very large, or, what amounts to the same thing, $\frac{1}{m}$ or $\frac{z}{m}$ is made very small. With $\frac{1}{m}=0$, eq. (I) takes the simpler form

$$A_0=A_i\left(1+\frac{3}{4}b+\frac{A_c}{A_i}\right). \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (II)$$

From this equation of "practical regulation coefficients" Mollier has given in the diagram, Fig. 296, for various assumed values of z (ignitions) and a (miss cycles), the ratios of $\frac{\delta}{\delta_0}$, under the assumption that $\frac{A_c}{A_i}=\frac{1}{4}$ in all cases. Line AB gives the ratios when $z=1$, which is the most favorable case, while line BC for $\frac{1}{m}=0$ (most unfavorable case), gives from eq. (II) the only values of practical use. Line BC is the line of reference not only for the condition that any small load fluctuations occur at con-

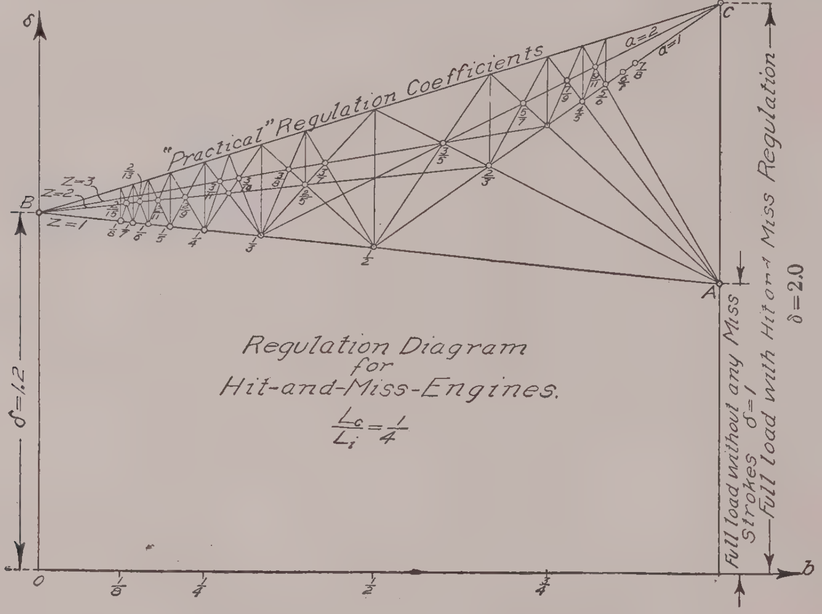

δ
"Practical" Regulation Coefficients
a=2
a=1
C
B
A
Z=3
Z=2
Z=1
δ=1.2
δ=2.0
Full load without any Miss Strokes δ=1
Full load with Hit and Miss Regulation
Regulation Diagram for Hit-and-Miss-Engines.
$\frac{L_c}{L_i}=\frac{1}{4}$
0
$\frac{1}{8}$
$\frac{1}{4}$
$\frac{1}{2}$
$\frac{3}{4}$
b

FIG. 296.—Diagram for Hit-and-Miss Regulation.

---

will always now and then occur no matter what the interval. Hence eq. (I) above instead of eq. (13) should be used for computing A for any load no matter how little below the ideal full load. For full load eq. (I) gives

$$A-A_i\left(1-\frac{1}{\infty}+\frac{3}{4}\cdot 1+\frac{A_c}{A_i}\right)=A_i\left(\frac{7}{4}+\frac{A_c}{A_i}\right),$$

which, with $\frac{A_c}{A_i}=\frac{1}{4}$, becomes $A=2A_i$. This also is indicated in Fig. 296 by the point C, and shows that as soon as the ideal condition of no-miss cycles at full load is departed from the original value of δ is increased very materially.

siderable intervals, but also when these fluctuations, grouped about the mean value of load, occur at intervals of any desired length. For this reason at $\frac{3}{4}$ load, for instance, the regulation coefficient may still be computed from eq. (II) if the governor acts as indicated in the series: $|||-||-|||-||-|||-||-$. Thus not only the effects due to load fluctuations, but also those due to any lag in the governor, are to a certain extent taken care of.

The line *BC*, Fig. 296, shows that the closeness of regulation in practice increases under decreasing loads.

The tachometer record, Fig. 297, shows with special clearness the magnitude and suddenness of speed changes sometimes occurring in hit-and-miss engines. At the end of the expansion stroke of the second cycle the number of revolutions per minute is 3% above, and at the end of the fourth, which is a miss cycle, it is 3% below the normal speed. Inside of these cycles, therefore, the crank will show a speed change of 6%,—and this with a fly-wheel weight of 22000 lbs., which in this case amounts to 220 lbs. for each nominal horse-power. It should be noted, however, that this record represents extraordinarily unfavorable operating conditions. The upper speed limit (103%) could only have been reached by a very heavy explosion, while the lower limit is in part at least due to the partial failure of ignition in the previous cycles. Under less unfavorable circumstances, the speed limits are much narrower than this, as is shown by Fig. 298, which is obtained from the same engine under the same load as for Fig. 296. If the power of an engine is transmitted by belting, the speed variations, owing to the equalizing action of the belt, will be much less pronounced in the power consumer, especially if this itself possesses rotating masses. Under such conditions there will be established in the power consumer an average speed for every four-stroke cycle, which is much closer to the normal speed than that measured in the prime mover itself. The advantage of equalization possessed by this elastic transmission is, however, lost if the speed increase or decrease in the engine extends over a considerable number of revolutions. This is clearly shown in the record of Fig. 298, for which the 100 H.P. engine was loaded only with the friction of a transmission about 100 ft. long. Here from 4–5 miss cycles are followed by 2 or 3 ignitions. During the former the average speed slowly decreases, while during the latter it increases by rapid stages. The maximum and average variations occurring under these conditions are well shown.

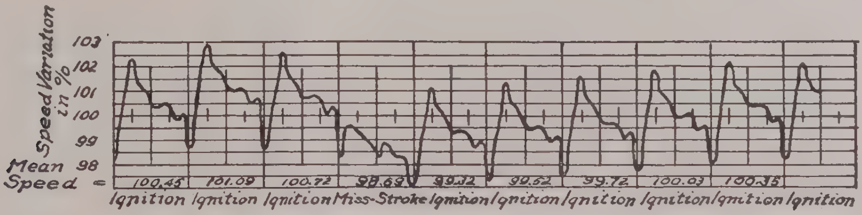


Fig. 297.—Tachogram, 100 H.P. Crossley Engine.

While it is true that speed fluctuations of the kind outlined do not seriously affect operation in ordinary industrial applications, they badly interfere with satisfactory electric lighting service and other operation requiring close regulation. To be on the safe side in cases of that kind, the designer has only one recourse: to make the fly-wheel, in the case of hit-and-miss regulation, at the outset twice as heavy as the computation (assuming a given value of δ for full load, i.e., without any miss cycles), seems to call for. Besides this, however, the greatest care should be taken not only in the design of the governing mechanism but also in the features of the mixing and ignition apparatus, in order to restrict the number of consecutive miss cycles to as small a number as possible and to insure that the first explosion following a period of retardation shall occur at the proper moment and be of full strength. But even under the best conditions, hit-and-miss regulation will require comparatively large rotating masses in order to satisfy the closer degrees of regulation demanded to-day at all loads on the engine.

As may be seen from the above discussion, the method of fly-wheel computation developed depends entirely upon the assumption that the small triangular areas $x+y$, which are a part of one quarter of the total uniform resistance (see Fig. 293) may be neglected in comparison with the positive work area A_a. In the case of the single cylinder 4-cycle and the old 6-cycle, this assumption is justified; but the case is different in engines in which the crank travel between successive ignitions is *shorter*. With any given engine power, N_i, the ordinate p_w of the uniform resistance varies inversely with the length b of the crank travel. The shorter this travel, the greater this ordinate [see eqs. (5) to (5*b*)], and owing to this fact the area ratio $\frac{x+y}{A_a}$ increases very rapidly, almost as the square.

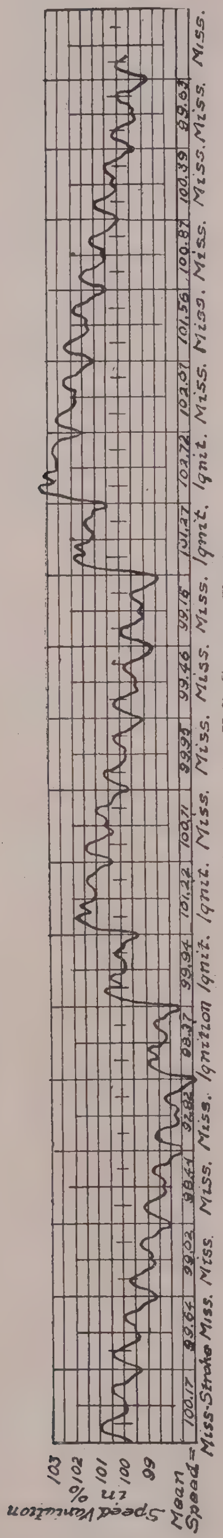


FIG. 298.—Tachogram, 100 H.P. Crossley Engine.

Thus, while in the single-cylinder 4-cycle ($\beta=720°$) this ratio does not usually reach .01, in the 2-cycle and the tandem or double 4-cycle ($\beta=360°$) it has already increased to from .03 to .04, and in the case of the double-acting 2-cycle or the double-acting two-cylinder 4-cycle ($\beta=180°$) it even reaches .15 to .16. Such quantities can of course be no longer neglected, and in order to take them into account eqs. (16), (17), or (16*a*) and (17*a*), require the following modification:

If, for example, a 4-cycle engine with a wheel rim weight of G lbs. is operated as a 2-cycle, the crank travel β between ignitions decreases from 720° to 360°, and hence a smaller weight of rim, equal to, say xG lbs., is sufficient to maintain the former degree of regulation. Further, making the engine operate as a 2-cycle doubles the previous capacity of the 4-cycle machine, so that for the same δ and the same N_i as before a rim weight of only $.5xG=\kappa G$ is really required. The factor x or κ has very different values depending upon the length of crank travel, i.e., upon the number of revolutions between ignitions, but is, on the other hand, little influenced by the shape of the indicator diagram, as was shown from a number of tangential effort diagrams. This fact makes it possible to regard κ as a constant for the same operating cycles and similar cylinder or crank combinations. Eqs. (16) to (17*a*) then take the following form:

$$G=\frac{\kappa\, 2125000\, N_i(.75+\rho)}{\delta V^2 n} \text{ lbs.}; \quad \ldots \ldots \quad (21)$$

$$\delta=\frac{\kappa\, 2125000\, N_i(.75+\rho)}{G V^2 n}; \quad \ldots \ldots \quad (22)$$

or

$$G=\frac{\kappa\, 77.5\times 10^7(.75+\rho)N_i}{\delta D_s^2 n^3} \text{ lbs.}; \quad \ldots \ldots \quad (21a)$$

$$\delta=\frac{\kappa\, 77.5\times 10^7(.75+\rho)N_i}{G D_s^2 n^3}. \quad \ldots \ldots \quad (22a)$$

For values of κ for the usual types of engines and operating cycles, see Table 22. The figures there given were determined from the tangential effort diagrams of Plates V, VI, and VII. In the construction of these diagrams, Figs. 5, 6, and 7 (Plate II), referring to a Körting engine for which $D=14.95''$, $S=26.2''$, and $n=$ 140 r.p.m., were used as the basis. The turning effort diagrams of Plates V, VI, and VII therefore strictly apply only to one engine for which the indicator diagram showed $\rho=.33$, but it is permissible, from what has been shown, to regard the values of κ in columns 5, 6, 10, and 11 of Table 22 as sufficiently accurate for all practical computations. For comparing the respective values of the coefficient κ the 4-cycle machine has been considered as the standard of reference. The values of κ for the other types and combinations show directly how much more favorable are the governing conditions in these cases. Columns 6 and 11, giving the values of κ_2 and κ_4 referred to the same engine capacity N_i, are the best to use in making direct comparisons, especially if it is desired to determine the saving in fly-wheel weight due to a given combination of cylinders.

The disadvantage of the single-acting single-cylinder 4-cycle engine with respect to fly-wheel weight necessary for a given engine capacity and coefficient of regulation is clearly apparent from columns 8 and 13 of the table.

For this type of engine, with $\delta=\frac{1}{40}$ and $V=65.8$ ft. per second, a fly-wheel weight (including weight of arms) of 148.5 lbs. per I.H.P. is required. To satisfy the same requirements in a single-acting 2-cycle cylinder requires only 59.6 lbs. per I.H.P.; in a double 2-cycle, according to combination No. IV or VI (see table), it requires only 12.5 lbs.; and in a three-cylinder 2-cycle only 5.94 lbs. per I.H.P. Compared to this last value, even the three-cylinder 4-cycle still requires 33.7 lbs. per I.H.P., that is, a fly-wheel weight about six times greater.

With the aid of double action, the coefficient of regulation in the single-cylinder, single-acting 4-cycle improves in the ratio of 148.5 to 91.3 (see Table 22, column 8), that is, by about 38%. That in itself is considerable, but is only one half the amount gained by making the single-cylinder 2-cycle double-acting (from Table 22, column 13, the gain is $=\frac{59.6-15.6}{59.6}=74\%$). In the building of large gas engines at present the double-acting 4-cycle and the single- or double-acting 2-cycle are the types of engines most often met in competition. Between these the following relations exist with respect to the coefficient of regulation, as taken from Table 22:

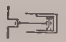
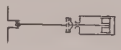
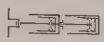
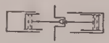
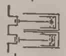
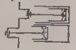
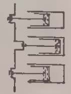
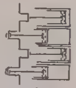
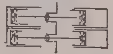
4-CYCLE.		2-CYCLE.	
Single Acting.	Double Acting.	Single Acting.	Double Acting.
1.000	.615	.401	.106
1.626	**1.000**	.652	.172
2.494	1.533	**1.000**	.254
9.434	5.802	3.783	**1.000**

Table 23 following, based on the preceding discussion, has been constructed for practical use. The quantities $.9\,G_i$ and GD_s^2 were determined from eqs. (16) and (21) or (16a) and (21a), using the assumed average values of κ, ρ, V, and n, as given. For any conditions not given in the table the desired results may easily be obtained directly from the proper equations. The constant C results from the contraction of the factors

$$\kappa\, 77.5\times 10^7(.75+\rho)=C, \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad (23)$$

TABLE 22

REGULATION FOR VARIOUS CYLINDER COMBINATIONS, ASSUMING $V=65.8$ FT. PER SEC., $\delta=\frac{1}{40}$, AND $\rho=.33$.

1	2	3	4	5	6	7	8	9	10	11	12	13
			4-CYCLE ENGINES.					2-CYCLE ENGINES.				
Number.	Cylinder and Crank Combination of the Various Types Considered.	No. of Cylinders.	Crank Travel β between Ignition in Degrees.	For Equal Dimensions, D, S and n. κ_1	For Equal Max. Capacity, N_i max. κ_2	N_i max. H.P.	$G_i=\frac{G}{N_i \text{ max.}}$ Lbs.	Crank Travel β between Ignition in Degrees.	Same, D, S and n. as in 4-cycle Engine, which is taken=1.0 κ_3	Same Capacity, N_i max. κ_4	N_i max. H P.	$G_i=\frac{G}{N_i \text{ max.}}$ Lbs.
I		1	720	1.000	1.000	65.8	148.5	360	.802	.401	131.6	59.6
II		1	540 180	1.230	.615	131.6	91.3	180	.424	.106	263.2	15.6
III		2	360	.796	.398	131.6	58.8	360	1.595	.399	263.2	53.8
IV		2	540 180	1.290	.645	131.6	95.6	180	.335	.084	263.2	12.5
V		2	360	.792	.396	131.6	58.6	360	1.602	.401	263.2	59.6
VI		2	540 180	1.290	.645	131.6	95.6	180	.335	.084	263.2	12.5
VII		3	240	.678	.226	197.4	33.7	120	.237	.0395	394.8	5.94
VIII		4	180	.335	.084	263.2	12.5					
IX		4	180	.335	.084	263.2	12.5					

For the double-crank 2-cylinder opposed engine ($\beta=360°$) the values of combination No. V, and for the 4-crank, 4-cylinder opposed engine ($\beta=180°$) the values of combinations Nos. VIII or IX are nearly exact.

from eq. (16*a*) or (21*a*) assuming $\rho=.35$, and taking the proper value of κ from Table 22. With this simplification the rim weight required for a given machine with $\rho=.35$ is then

$$G=\frac{CN_i}{\delta D_s^2 n^3} \text{ lbs.}, \qquad (24)$$

and the moment is

$$GD_s^2=\frac{CN_i}{\delta n^3} \text{ ft.-lbs.} \qquad (24a)$$

Hence

$$\frac{\delta}{C}=\frac{N_i}{GD_s^2 n^3}. \qquad (25)$$

Equations similar to the above, involving a constant C, are well known in steam-engine design. The mean rim velocity $V=65.8$ ft. per sec., upon which the results of Table 23 are based, is good for horse-powers up to about 15 or 20. V increases gradually with engine capacity to the limiting value of from 120 to 130 ft. per sec. (see p. 234). As will be seen from eq. (21), since V enters as the square, any variation in its value will have a marked effect upon the value of G. Hence the importance of the proper choice of rim velocity is apparent. The set of curves on Plate VII, the use of which is there explained, will be found of great utility in this respect.

TABLE 23

REQUIRED FLY-WHEEL RIM WEIGHT AND MOMENT GD_s^2 FOR $V=65.8$ FEET PER SEC., $n=200$ R.P.M., $\delta=\frac{1}{40}$, $\rho=.35$ AND κ FROM TABLE 22.

1	2	3	4	5	6	7	8	9	10
		4-cycle Engines.				2-cycle Engines.			
Number.	Number of Cylinders and Cranks.	Crank Travel β, Degrees.	Rim Weight $.9G_i$ required per I.H.P., Pounds.	GD^2_s required per I.H.P., Ft.-lbs.	Constant C from Eq. (23).	Crank Travel β, Degrees.	Rim Weight $.9G_i$ required per I.H.P., Pounds.	GD^2_s required per I.H.P., Ft.-lbs.	Constant C from Eq. (23).
	Single-acting Types.								
I	1 cyl., 1 crank	720	133.5	4265	85.3×10^7	360	53.7	1710	34.2×10^7
II	2 cyls., 1 crank	540 180	86.2	2745	54.9×10^7	....	....	....	....
III	2 cyls., 2 cranks	360	53.0	1695	33.9×10^7	180	11.2	358	7.16×10^7
IV	3 cyls., 3 cranks	240	30.1	962	19.25×10^7	....	....	....	....
V	4 cyls., 2 cranks	180	11.2	358	7.16×10^7	....	....	....	....
VI	4 cyls., 4 cranks	180	11.2	358	7.16×10^7	....	....	....	....
	Double-acting Types.								
VII	1 cyl., 1 crank	540 180	82.3	2620	52.4×10^7	180	14.2	452	9.04×10^7
VIII	2 cyls., 1 crank	180	11.2	358	7.16×10^7	....	....	....	....
IX	2 cyls., 2 cranks	180	11.2	358	7.16×10^7	90	2.0	68.5	1.27×10^7
X	4 cyls., 2 cranks	90	2.29	71.5	1.43×10^7	....	....	....	....

Under normal conditions of operation fly-wheels are commonly designed for the following coefficients of regulation:

For ordinary power purposes and similar installations .... $\delta=\frac{1}{30}$ to $\frac{1}{40}$
For electric lighting service (without storage batteries) with
direct-current generators .... $\delta=\frac{1}{70}$ to $\frac{1}{80}$
Alternating-current generators .... $\delta=\frac{1}{125}$ to $\frac{1}{150}$

In case of operation of generators in parallel[1] and for direct connected generators, these values of δ should be further decreased by from 30 to 60%.

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[1] For thorough discussions of this question, see Zeitschrift d. V. D. I., 1904, p. 793, and Electrotechnische Zeitschrift, 1902, No. 14.

2. Determination of Dimensions.

Fly-wheel Designs. (See also the examples shown in the Plates of Part III.)

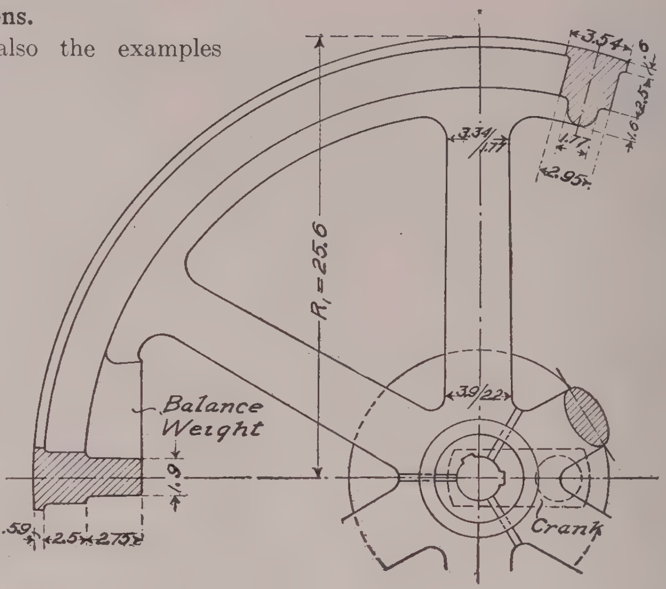


Fig. 299.—Fly-wheel, 6 H.P. Bánki Engine, Ganz & Co., Budapest.

$D = 5.5''$,

$S = 10.0''$,

$n = 300$ r.p m.

(The projecting balance weight destroys the symmetry of the wheel and bothers while starting. The necessity for such weight should therefore be avoided.)

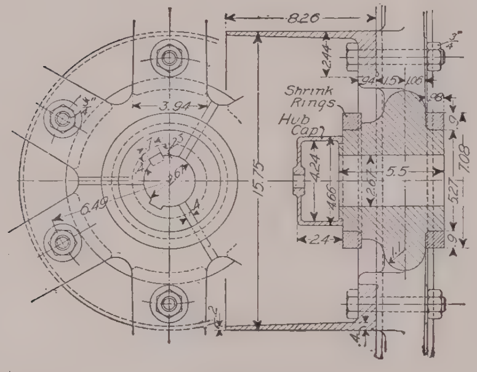


Fig. 300.—Hub and Belt Pulley for Wheel shown in Fig. 299.

(The purpose of dividing the hub is to avoid casting stresses in the arms. The joints are filled in and the shrink rings put into place before the hub is bored.)

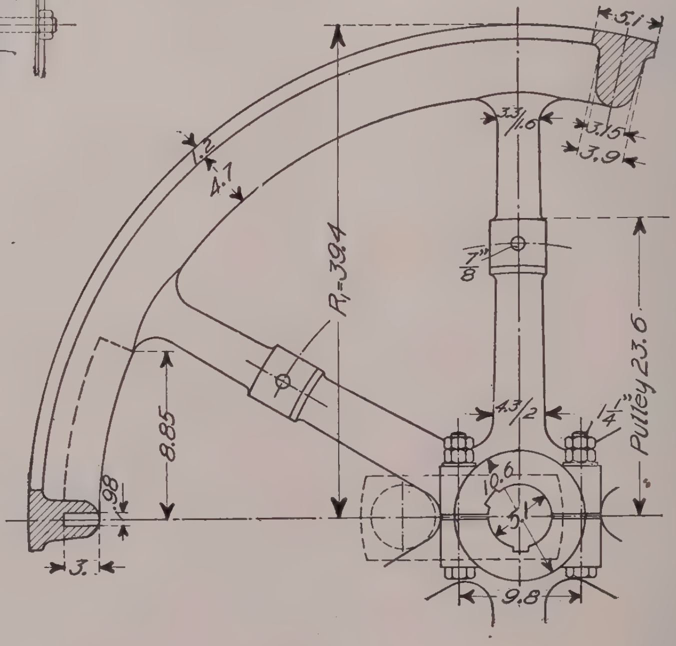


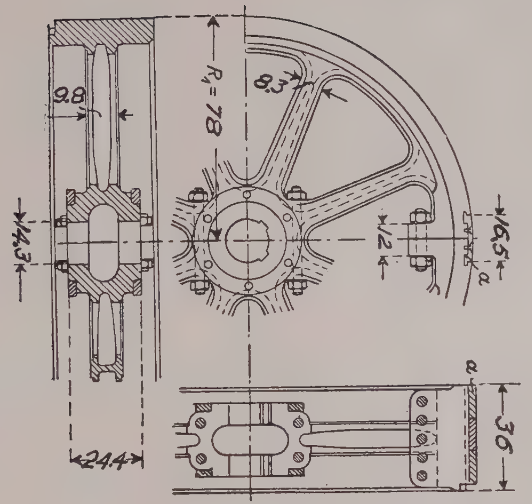
Fig. 301.—Fly-wheel, 25 H.P. Wenzel Engine, Fr. Zimmermann A.-G., Halle S.

$D = 13.4''$,

$S = 18.9''$,

$n = 160$.

(The cutting away of metal in that part of the rim on the same side with the crank serves to help balance the rotating parts. Note the strengthening of the arms where the belt pulley is attached.)



FIGS. 302–304.—Fly-wheel for a 400 H.P. Crossley Engine.

On account of the unusual width of face, the arms are made I section. Rim connection is made with 5 bolts on the inside and 3 countersunk links (*a*) on the outside.

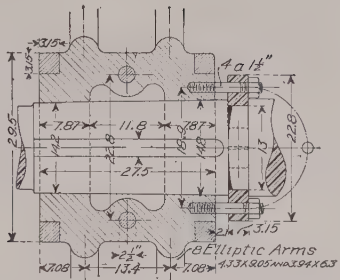


FIG. 305.—Hub of a large Armature Fly-wheel.

Diameter of wheel 14.75 ft., face 24″. Wheel is drawn into place by means of four 1½″ screws which are held in a two-part ring let into a groove in the shaft. This is for the purpose of insuring proper seating and true running.

Constructive Details. The static computation of fly-wheels and especially the check computation regarding the stress relations chosen is possible only on the basis of a number of more or less arbitrary assumptions, since neither the magnitude of the stresses occurring nor the method of their action and distribution in the wheel can be definitely determined. To be on the safe side, the *most unfavorable* set of conditions among those that may occur should be chosen, and the dimensions determined accordingly. Besides this, the allowable stress, especially in the arms, should be kept low. In this respect it should be noted that the stress in the arms, due to the constant energy exchange between wheel rim and crank, becomes a reversing stress, varying from a positive to a negative maximum value.

The weight of a wheel rim having a cross-section of f sq.in. and a radius of R ft. to the center of gravity of the rim cross-section may be expressed by

$$G=\frac{2\ R\pi f\ 450}{144}=19.63\ Rf \text{ lbs.}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (26)$$

in which 450 lbs. is the weight of cast iron per cubic foot. A given weight, G lbs., therefore requires a cross-sectional area in the rim of

$$f=\frac{G}{19.63\ R} \text{ sq.in.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (27)$$

The diameter D_s of fly-wheels, to the center of gravity of the rim cross-section, is usually taken at from 4 to 5 S, or $R=2$ to $2.5\ S$. Large engines, however, sometimes compel the use of $R=3$ to $3.5\ S$. The maximum value of D_s depends upon the allowable value of the rim speed, which may be found from

$$V_{\max}=\sqrt{\frac{2.31\ gK_z}{\gamma}} \text{ ft. per sec.}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (28)$$

in which K_z may be taken at 1700 lbs. per sq.in. for cast iron, and γ=specific gravity of cast iron=7.25. The allowable velocity therefore is

$$V_{\max}=\sqrt{\frac{2.31\times32.2\times1700}{7.25}}=132.0 \text{ ft. per sec.}$$

This limiting velocity, however, should be approached only in extreme cases, because the fly-wheel rim is not only put under a tensile stress equal to

$$\sigma_z=.0972V^2 \text{ lbs. per sq.in.,} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (29)$$

as eq. (28) assumes, but it is subject also to considerable bending stresses due to the action of the arms. Hence the value $V=100–110$ ft. per sec. should not usually be exceeded.

The number of arms used in cast-iron wheels is usually six, unless the wheels exceed 10 ft. in diameter, when eight are employed. For widths of wheel face greater than 15 in., either arms of special cross-section or double spiders should be used. The dimensions of the arms depend, on the one hand, upon the bending moment due to the tangential force P (Fig. 306), and, on the other, on the tensile stress induced by the centrifugal force in the rim. The bending moment does not reach its maximum value owing to the pull of belt or rope, but owing to the inertia of the wheel rim should the engine be suddenly started or stopped. If we make the assumption[1] that at the moment of starting the maximum crank effort $T_{\max}$ occurs when the wheel is standing still ($V=0$), a condition strongly improbable and with certainty always preventable, the inertia of the wheel rim will cause at the hub end of each one of the arms (i in number) a bending moment equal to

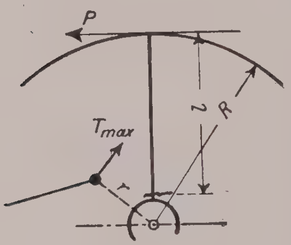


FIG. 306.

$$M_b=\frac{T_{\max}r}{iR}l \text{ in.-lbs.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (30)$$

The bending stress then will be

$$\sigma_b=\frac{M_b}{W}=\frac{M_b}{.0982a^2b}\sim\frac{10\,M_b}{a^2b} \text{ lbs. per sq.in.,} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (31)$$

in which a is the major and b the minor axis of the elliptical cross-section of the arms at the hub end. It is also here assumed that the radius of the center of gravity of the rim section belonging to each arm is R instead of R_0. In reality $R_0=.94$ to $.96\,R$, that is, R_0 is from 4 to 6% smaller than R.

---

[1] It would seem at first sight that the bending moment on the arms would be a maximum when, under full velocity ($V_{\max}$), owing to pre-ignition, the crank suddenly receives an impulse in a direction *opposed* to the direction of rotation. In actual operation this is not the case, since early ignitions with full development of explosion pressure (P_z) can only occur near the dead center. But the turning moment produced by them in these positions is always less than the bending moments due to the maximum tangential effort.

Too great tapering of the arms should be avoided; a good rule to follow is to decrease a by $.18\,R$ in., and b by $.10\,R$ in. for the rim cross-section of the arms, where R represents the radius of the wheel in feet.

The centrifugal force due to the part of the rim belonging to each arm is

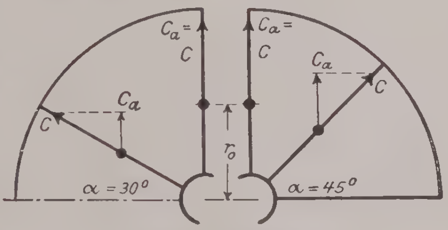
$$C=\frac{MV^2}{iR}=\frac{GV^2}{giR}=\frac{.031\,GV^2}{iR}=\frac{.00034\,GRn^2}{i}\text{ lbs.} \qquad (32)$$

This force tends to rupture the arm in tension at its weakest place. The resulting tensile stress is $\sigma=\frac{C}{f_a}$ lbs. per sq.in., which for elliptical arms is

$$\sigma=\frac{C}{.7854ab}\text{ lbs. per sq.in.} \qquad (33)$$

The maximum tensile stress usually occurs at the outer ends of the arms. If, however, the arms should be weakened in any way, as for instance, by the drilling of holes for the bolts fastening the belt pulley, a check computation of σ_b and σ is to be carried through for these cross-sections.[1]

The connections of split wheels are subject mainly to the disruptive action of the centrifugal forces acting normally to the plane of division. It is important therefore that these fastenings take up this stress as directly as possible, i.e., without setting up any considerable bending moments. Since the load produced by these forces is never equally distributed over the rim and hub connections, and since there is no means of determining what this distribution is in any given case, the usual method is to make each of these connections strong enough to safely withstand by itself the centrifugal forces on one-half of the wheel. Shrink rings or similar means should not be depended upon to any extent because, even with the wheel at rest, they may be already stressed to the elastic limit.



FIGS. 307 and 308.

The sum of the centrifugal forces acting on the connections of a divided wheel consists of those (C_k) due to half the wheel rim and those (C_a) due to the arms of one half of the wheel. Both of these should be referred to the plane of division and considered as acting at their respective centers of gravity. Thus, with reference to eq. (32) and Figs. 307 and 308,

$$C_k=.00034\,Rn^2\frac{G}{2}\times\frac{2}{\pi}=.00011\,Rn^2G\text{ lbs.,} \qquad (34)$$

and for each arm of weight G_a,

$$C_a=.00034\,G_ar_0n^2\sin\alpha\text{ lbs.} \qquad (35)$$

---

[1] TRANSLATOR'S NOTE. In many cases designers also check the hub end of the areas for tension due to centrifugal force and add this tensile stress to the bending stress as found from eq. (31) to find the total maximum stress.

For the middle arm $\sin\alpha=\sin 90°=1.0$. For the two side arms $\sin\alpha=\sin 30°=.5$ when $i=6$, and $\sin\alpha=\sin 45°=.707$, when $i=8$. The total centrifugal force due to the three arms will then be, referred to the plane of division, when $i=6$,

$$C_a=.00034\ G_a r_0 n^2(1+2\times .5)=.00068 G_a r_0 n^2 \text{ lbs.} \quad \ldots\ldots \quad (35a)$$

and when $i=8$,

$$C_a=.00034\ G_a r_0 n^2(1+2\times .707)=.00082\ G_a r_0 n^2 \text{ lbs.} \quad \ldots\ldots \quad (35b)$$

The sum of the forces acting on the connections will therefore be

$$\Sigma(C)=C_k+C_a \text{ lbs.} \quad \ldots\ldots \quad (36)$$

The method of computing the stresses existing in the connections will be shown by the following numerical example. This shows that under certain circumstances these stresses may be quite considerable, and in properly taking care of them the designer is limited by the fact that for large values of R or V the joint should be made as light as possible to avoid additional centrifugal stresses. If this cannot be readily done, it is better to cast the rim in one piece and divide the spider only.

No computation is possible for the hub. Experience has shown that the following dimensions are satisfactory:

Length of hub $=1.5$ to $2.5\,d$ (the latter value only in small engines) and external diameter $=2$ to $2.5\,d$, where $d=$ shaft diameter.

It is usual to core out the middle of the hub for about one third of its length.

Example. Computation for a split wheel for an engine having a diameter of cylinder $D=17.7''$, stroke $S=23.6''$, $n=160$ r.p.m., coefficient of regulation $\delta=\frac{1}{40}$. The tangential effort diagram shows an excess area equivalent to 54 200 ft.-lbs. of work.

Then from eq. (9), p. 218, the necessary weight of rim will be

$$G=\frac{2900\,A}{\delta(Rn)^2}=\frac{2900\times 54\,200}{\frac{1}{40}(R\times 160)^2} \text{ lbs.}$$

Making $R=2.5\,S=2.5\times 23.6''=4.92$ ft., which corresponds to $V=82.6$ ft. per sec., we have

$$G=\frac{2900\times 54\,200}{\frac{1}{40}(4.92\times 160)^2}=10\,200 \text{ lbs.}$$

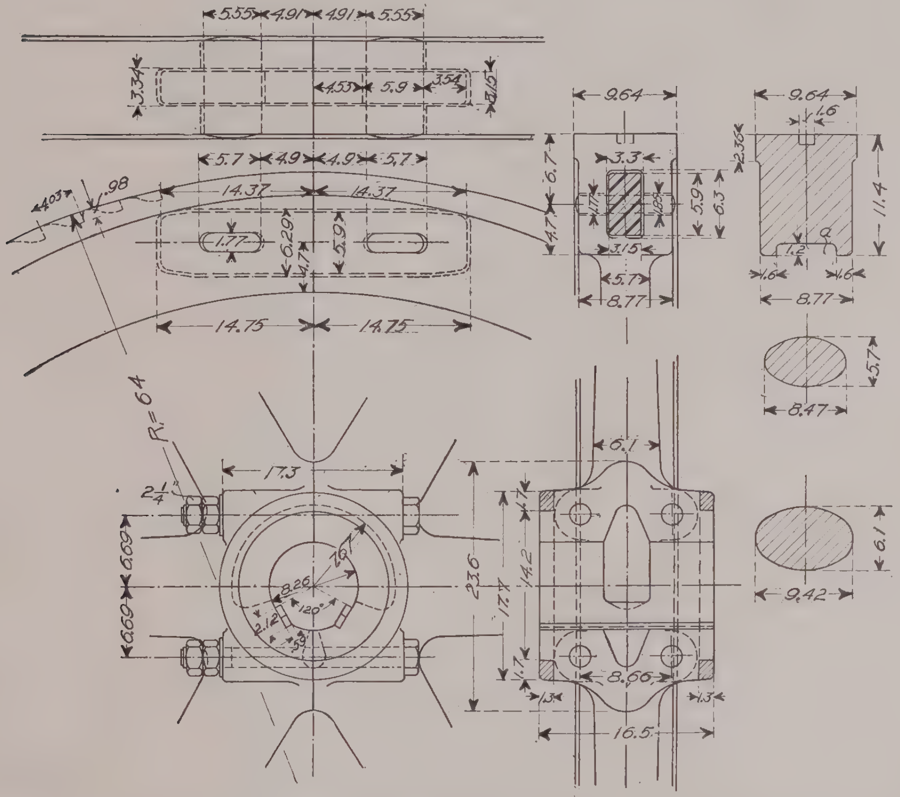
Assuming that one tenth of this weight is due to the arms, only $\frac{9}{10}\times 10200=9180$ lbs. needs to be furnished in the rim.

The cross-sectional area of rim required for this weight is from eq. (27), p. 233,

$$f=\frac{G}{19.63\,R}=\frac{9180}{19.63\times 4.92}=95.5 \text{ sq.in.}$$

This has been amply provided for in Fig. 314. After laying down the dimensions of the hub according to the empirical data above given, we proceed to compute the stresses in the arms. With the dimensions given in Figs. 309–316, we find these as follows:

$$T_{\max} = .5\ P_z = .5 \times 280\ D^2 = .5 \times 280 \times (17.7)^2 = 44000 \text{ lbs.}$$



Figs. 309–316.

(Recess *a* may be filled with lead when balancing the wheel. The two parts of the rim lying in the plane of divisions are not thus cut away.)

Maximum bending moment at the hub cross-section of arm therefore is, from eq. (30)

$$M_b = \frac{44\,000 \times .985}{6 \times 4.92} \times 49.5 = 72\,800 \text{ in.-lbs.},$$

from which

$$\sigma_b = \frac{10\ M_b}{a^2 b} = \frac{10 \times 72\,800}{(9.42)^2 \times 6.1} = 1340 \text{ lbs. per sq.in.},$$

which may be considered safe.

The centrifugal force of one-sixth of the rim is

$$C = \frac{.00034\ GRn^2}{i} = \frac{.00034 \times 9180 \times 4.92 \times (160)^2}{6} = 65\,000 \text{ lbs.}$$

The tensile stress induced in the cross-section of the arms near the rim by this force is

$$\sigma=\frac{65000}{.7854\times 8.47\times 5.7}=1750 \text{ lbs. per sq.in.},$$

which, in view of the extreme assumptions made, may also be considered safe.

To obtain the stresses in the joints we proceed as follows: The weight G_a of each arm is about 550 lbs., and $r_0=2.46$ ft. Hence, according to eqs. (34) and (35a),

$$C_k=.00011\ Rn^2G=.00011\times 4.92\times 160^2\times 9180=127\,000 \text{ lbs.},$$

$$C_a=.00068\ G_a r_0 n^2=.00068\times 550\times 2.46\times 160^2=23\,600 \text{ lbs.};$$

$$\Sigma(C)=127\,000+23\,600=150\,600 \text{ lbs.}$$

Assuming the most unfavorable condition, i.e., neglecting the shrink rings and the rim connection altogether, each of the four $2\frac{1}{4}''$ hub bolts will have to carry $\frac{150\,600}{4}=37\,700$ lbs. This causes a tensile stress over the cross-section at the bottom of the thread of $\frac{37\,700}{2.93}=$ 12 900 lbs. per sq.in. The real stress is of course considerably less than this.

The two rim connections, with the same unfavorable assumption, are each loaded with $\frac{150\,600}{2}=75\,300$ lbs.

Key. This is under a bending moment

$$M_b=\frac{75\,300}{4}\times 3.34=62\,900 \text{ in.-lbs.};$$

hence the bending stress

$$\sigma_b=\frac{62\,900\times 6}{1.77\times(5.55)^2}=7000 \text{ lbs. per sq.in.}$$

There is also a shearing stress equal to

$$\tau=\frac{75\,300}{2\times 1.77\times 5.55}=3850 \text{ lbs. per sq.in.}$$

With a stress ratio of $\alpha_0=\frac{K_b}{1.3\,K_s}\sim 1.0$ (for soft steel), the resultant stress will then be, in the most unfavorable case,

$$\sigma_r=.35\times 7000+.65\sqrt{7000^2+4(3850)^2}=9200 \text{ lbs. per sq.in.}$$

Link. This is subject to a bending moment

$$M_b=\frac{73\,500}{8}\times 1.85=17000 \text{ lbs. per sq.in.},$$

from which bending stress

$$\sigma_b=\frac{17000 \times 6}{3.15\times(3.54)^2}=2600 \text{ lbs. per sq.in.}$$

The shearing stress also occurring is

$$\tau=\frac{75300}{2\times3.15\times3.54}=3390 \text{ lbs. per sq.in.}$$

Both of these stresses are so low that it is not worth while to combine them.

Ends of the Rim. The bending moment is

$$M_b=\frac{37650}{8}\times1.85=8700 \text{ lbs. per sq.in.}$$

and

$$\sigma_b=\frac{8700\times6}{\frac{8.77-3.3}{2}\times(4.91)^2}=790 \text{ lbs. per sq.in.}$$

The shearing force, considering it as single shear distributed in this case over two surfaces, each $4.91''\times\frac{8.77''-3.3''}{2}$ is

$$\tau=\frac{75300}{2\times 4.91\left(\frac{8.77-3.3}{2}\right)}=2800 \text{ lbs. per sq.in.}$$

For the case in hand (cast iron, rectangular cross-section) α_0 is approximately $\frac{1.7}{1.3}=1.3$, hence the resultant stress is

$$\sigma_r=.35\times790+.65\sqrt{790^2+4(1.3\times2800)^2}=5000 \text{ lbs. per sq.in.},$$

which, in view of the very unfavorable assumptions made, is perhaps just safe. The pressure between rim and key is

$$k=\frac{37650}{1.77\times2.73}=7800 \text{ lbs. per sq.in.}$$

That between link and key is

$$k=\frac{75300}{1.77\times3.15}=13500 \text{ lbs. per sq.in.}$$

X. Governors

1. Methods of Governing. Neglecting certain methods of regulating the speed of automobile engines, consisting in varying the point of ignition, entire suppression of the spark, varying the size of the compression space, etc., the methods of governing 4-cycle engines may be divided into three fundamentally different groups:

(*a*) *Hit-and-miss Governing* (ratio of air to gas and quantity of charge per cycle remain unchanged). This method may be carried out in any of the following ways:

Keeping the fuel valve closed (operation of inlet and exhaust valves normal).

Holding the exhaust valve open (automatic inlet valve remains closed).

Keeping the inlet valve closed (operation of exhaust valve normal, decided vacuum during suction stroke).

(*b*) *Governing by Changing the Quality of the Charge,* diagram Fig. 317 (quantity of charge constant and ignition regularly every fourth stroke).

This may be carried out by:

Changing the lift or duration of opening of the fuel valve, or changing the lift or duration of opening of the air inlet valve (the fuel valve being automatic); or

Drawing back burned gases to dilute the charge (gas and air valves, or inlet valve, being automatic).

(*c*) *Governing by Changing the Quantity of Charge,* diagram Fig. 318. (Fuel mixture of constant composition, ignition every fourth stroke.) This may be carried out by:

Forcing a part of the combustible charge back into the suction mains; or

Varying the moment of closing the inlet valve; or

Throttling the charge during the entire suction stroke.

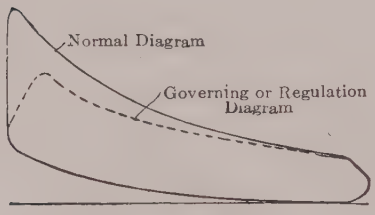


Fig. 317.

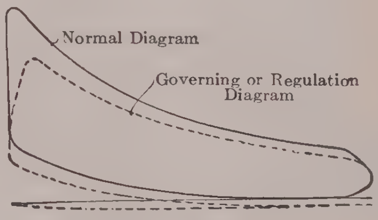


Fig. 318.

Two-cycle engines of small capacity and also those using liquid fuels may be regulated according to method (*a*) by temporarily cutting out the gas or oil pump. For the larger 2-cycle machines method (*b*) is commonly employed, adjusting the quantity of fuel gas furnished to the cylinder by changing the delivery of the pump (by means of throttling the suction, forcing back a part of the charge into the suction mains, etc.). In this case, if the pump does not force its charge directly into the cylinder but into an intermediate receiver, there is apt to be a lag of two or three revolutions before any given governor action commences to affect the power generated.

The complete application of method (*c*) to scavenging 2-cycle engines fails on account of the fact that even at the minimum loads the cylinder must be completely filled with air, which can be only partly forced back again. Methods (*b*) and (*c*) may, however, be used in combination in 2-cycle engines having inlet valves (not merely inlet ports controlled by the piston) by using (*c*) for the upper ranges of load, forcing a part of the mixture out of the cylinder, and using (*b*) for the lower loads, governing by regulating the quantity of fuel gas.

In general, for 4-cycle engines using gas or gasoline, the hit-and-miss method of regulation is most economical, because it permits of the retention at all loads of the most favorable mixture and the most efficient maximum degree of compression. The method, however, possesses the disadvantage that with varying loads there is a considerable variation in speed (see p. 228), and it should therefore be used, especially for the larger engines, only when the coefficient of regulation may be as high as $\delta \geqq \frac{1}{40}$.

In the case of kerosene and alcohol engines, hit-and-miss regulation is usually less efficient because, owing to continued miss strokes, the temperature of the cylinder and vaporizing chamber may decrease to such an extent as to lead to a condensation of the fuel vapor on the walls, thus causing not only a fuel loss but also fouling the engine.

Methods (*b*) and (*c*) are usually considered "precision" methods of governing. In this respect the third method is somewhat better than the second, because in this the mean negative pressure (compression stroke) increases or decreases simultaneously with the mean positive pressure (expansion stroke), which in turn affects the variation of the tangential turning effort in a favorable sense. On thermal grounds (*b*) is preferable to (*c*) on account of the fact that the former at all loads works with constant compression, while the latter operates with decreasing compression under decreasing loads, and hence at smaller thermal efficiency.

The advantage of the higher compression in the second method is, however, considerably overbalanced by practical disadvantages. The lean gas mixtures incident to the lower loads are in themselves hard to ignite, and on that account burn only slowly and incompletely. The heat losses due to this state of affairs are further emphasized when the gas content of the charge is regulated by controlling the time of opening of the gas valve. Owing to the shortening of the time available for diffusion, the naturally lean mixture is but imperfectly formed and the result is incomplete combustion (see Part V). For these reasons method (*c*) is therefore generally better than (*b*) also on economic grounds. Other advantages of the third method are that it may be carried out with much simpler mechanical means (throttle valve); that with decreasing compression the frictional resistances also decrease; and that, owing to the high vacuum in the cylinder during the suction stroke, the lubricating oil is more apt to reach the less accessible parts of the piston, which of course betters the mechanical efficiency.

The system of speed regulation introduced by Letombe is in a certain sense a combination of methods (*b*) and (*c*). In this system the degree of compression is changed at the same time with the gas content of the charge, but in the opposite sense. Accordingly the leanest mixture is under the highest compression, which latter may be forced correspondingly high without danger of pre-ignition. On the other hand, although the richest mixture is under least compression, the ratio of expansion is increased, thus drawing down the otherwise unfavorably high terminal pressure and temperature (see description of engine in Part IV).

Several attempts have also been made at various times to carry out the third method of governing with constant compression ratio,—an idea which on thermal grounds is quite correct. This scheme requires that the compression space must change with the load in order to keep the compression pressure at the same point. These attempts have as yet not resulted in any successful constructions, although one of the older automobile engines worked on this principle for some time.[1]

In order to maintain a constant thermal efficiency for all fuel ratios it is not only necessary to regulate the degree of compression, but the point of ignition should also be adjusted at the same time to correspond with the particular gas content of the charge. Such adjustment, when attempted at all, has so far only been made when disturbances in the operation of the machine showed that something was wrong, and in such cases only by hand. Lately, however, attempts have been made to put the time of ignition under governor control. It must be admitted that such automatic regulation of the time of ignition is very difficult to properly carry out owing to accidents of operation. On the contrary, hand regulation of the spark serves its purpose very well and is consequently coming into more and more extended use for the larger machines.

2. Construction of Governors. Hit-and-miss regulation does not demand any great refinement in the governors used to carry it out. In them, power, stability, and range of movement are of almost no importance. It is necessary merely to cause the machine members employed to periodically interrupt the charging action to assume one or the other of two positions, which action usually does not call for any considerable expenditure of energy or constancy in the operation of the governor. For this reason this type of speed regulation may be carried out by simple swinging or oscillating members, which, having a certain freedom of movement, are forced out of their normal position by their own inertia. Of this type are the so-called

Pendulum Governors.

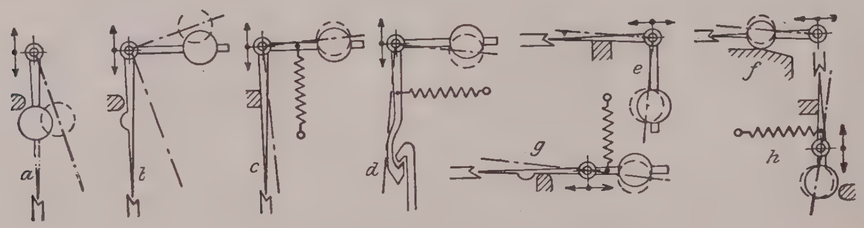


Fig. 319.—Fundamental Types of Pendulum Governors.

Of the fundamental forms of this type of governor shown in Fig. 319, form *a* shows the simple hanging pick blade, *b* the hanging bell-crank pick blade, and *c* the guided bell-crank pendulum. In all three forms the blade acts downward. Form *d* illustrates the bell-crank pendulum with hook latch arrangement working upward; *e* the guided bell-crank pendulum with horizontal blade; *f* the simple guided horizontal pick blade; *g* the double armed horizontal pick blade, and *h* the simple hanging pick blade governor working upward. Simple pick blades, just before their engagement with their respective valve stems, are thrown out of their normal position by interfering with fixed contact pieces, and are brought back into normal position at constant speed either by their own inertia or by spring pressure. Guided pendulums, on the other hand, are thrown out of their normal or engagement position by their own inertia only after $N_{\max}$ has been exceeded and their deflection is consequently considerably less than in simple pick blade governors.

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[1] See Güldner, Fahrzeugmotoren für flüssige Brennstoffe, p. 34.

Designs of Pendulum Governors:

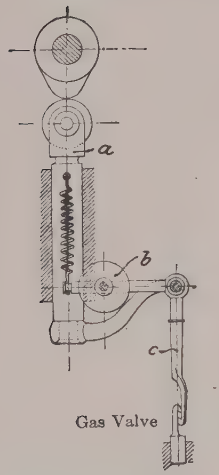


FIG. 320.—Gasmotoren-Fabrik, Deutz.

(At max. speed the weighted pendulum b, which is fastened to the valve rod a, lays so far behind that on the upstroke c fails to latch.)

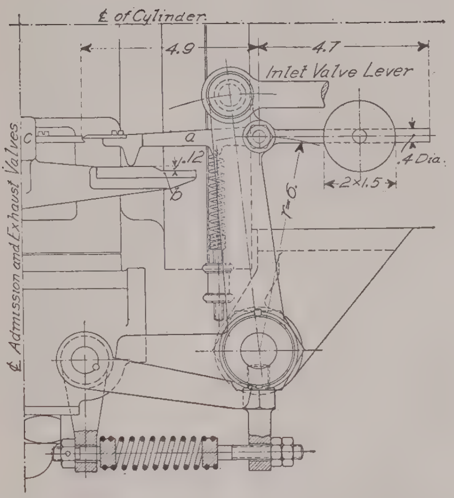


FIG. 321.—Pendulum Governor, Krupp-Grusonwerk.

(Blade a, at $n_{\max}$, is thrown above block c by the action of wedge b See Figs. 238 and 239.)

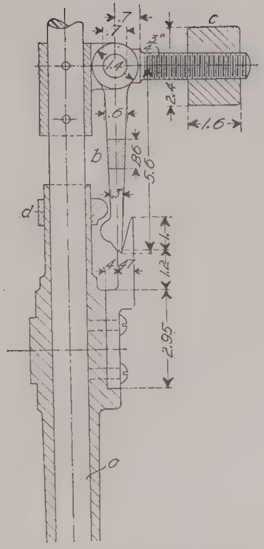
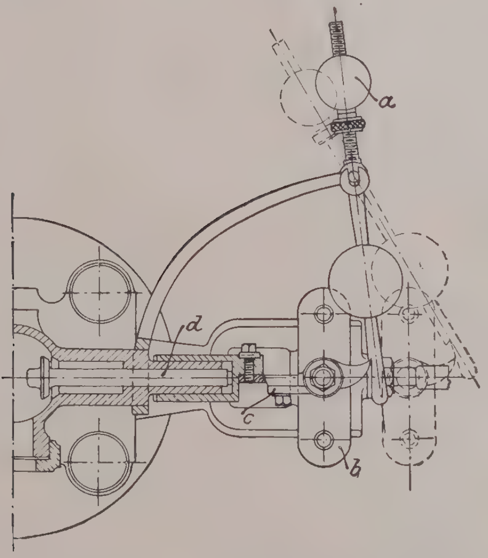


FIG. 322.—Pendulum Governor, Wenzel-Zimmermann.

(The pendulum b, carried by the exhaust valve rod a, prevents the closing of the valve when $n_{\max}$ is reached. Counterweight c and deflector d are adjustable.)



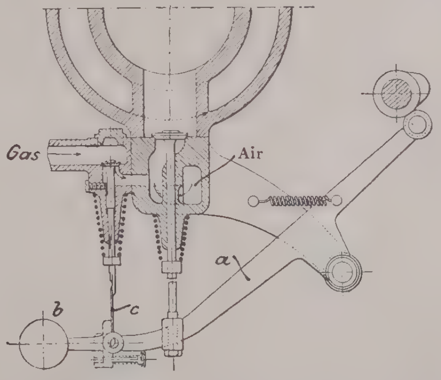


FIG. 323.—Pendulum Governor, Crossley Bros.

(Pendulum b, carried by valve lever a, at $n_{\max}$, lags behind far enough to cause blade c to miss the valve stem.)

FIG. 324.—Pendulum Governor, Delamare-Deboutteville, old Type.

(Pendulum a is carried back and forth by b. At $n_{\max}$ the notch at the lower end of a fails to engage the right hand end of the blade c. The left end of the latter drops down and misses the stem d. Fig. 237 shows another construction due to the same designers. Here the pendulum f is moved up and down with the valve lever $d'e$. The air pressure produced in the dash pot h deflects the vertical blade to the right, and at $n_{\max}$ causes it to miss the stem of the gas valve b altogether. Latch g holds f in this position until the former is unhooked in the highest position of b. The speed may be controlled by adjusting the small escape valve i.)

Details of Construction. To base the design of pendulum governors upon the laws of motion applying to the case does not offer a satisfactory solution and does not obviate subsequent trying out.[1] For that reason it is safer to depend upon judgment and feel and to provide, at least in the first models built, sufficient range of adjustment in those details of the construction upon which the action of the governor depends (length and weight of pendulum, strength of spring, angle of deflection, etc.). In any case, the tendency among German designers is to use the pendulum governor only for the smaller sizes of engine; for the larger machines they justly regard it as too largely subject to accidental influences in its regular operation. Delamare-Deboutteville perhaps went further than any other designer in the application of this type of governor, applying it even to his older 1000 H.P. Simplex engine.

The fact that hit-and-miss regulation calls for but a very small expenditure of energy, has led to the invention of the direct acting electrical hit-and-miss governor, an example of which is shown in Fig. 325. In this construction a centrifugal governor, when the speed exceeds the normal, simply closes the excitation circuit of a small electro-magnet *a*. This in turn attracts the pick blade *b*, pulling it out of reach of the valve lever *c*, and the gas valve *d* remains closed. The governor itself thus expends almost no energy.

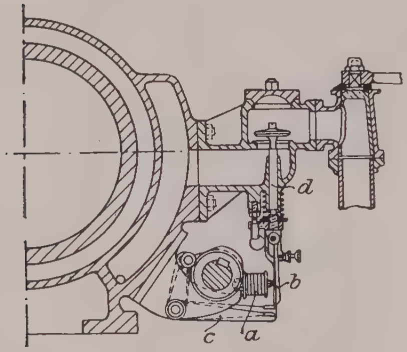


FIG. 325.—Kilmarnock's Electric Governor.

For the "precision" methods of governing, only *Centrifugal Governors* need be discussed, and, in general, the first thing to consider in their design is the range of movement of the sleeve or collar, power (energy) of the governor being a secondary item. Numerical constants for power necessary, which will be generally applicable, cannot be given on account of the variety of the constructions used; a plain throttle valve, for instance, usually requires less power in the governor than a movable cam, but more than a valve roller working on a fixed cam. It therefore becomes necessary either to measure the power required to move the governing mechanism and valves, or to estimate it. In the latter case the estimate should be at least very liberal.

In the case of all gases not absolutely clean, especially true of suction gases, the governing mechanism should be so constructed that a fouling of the controlling valve does not interfere with the governor operation. Butterfly valves, slides, and similar means which give good service when used with illuminating gas, soon cause difficulties when used with dirty gas in that deposits of tar or other incrustations seriously increase friction losses and finally prevent governor operation altogether. Such occurrences may be avoided by making the governor act not directly upon the controlling valve itself but upon that part of the valve gear operating this valve. In this case the valve exposed to the gas is positively operated by the gear and not by the governor. See examples of this construction in Figs. 326–329, also p. 245, etc.

The design of the centrifugal governors proper follows the general theory. It will, however, hardly be necessary here to enter into any exposition of the principles involved, especially since the engine builder makes his own centrifugal governors only when compelled to do so. For, unless peculiarities of construction in the engine make the use of stock governors found in the market impossible, it is cheaper to buy governors of equally good grade from builders making their manufacture a specialty.[1] In general, it is quite easy to avoid special construction. An exception to the rule we find only in the smaller machines where, on account of questions of low cost and the fact that close regulation is not a rigid requirement, only the very simplest governing mechanisms are used. For these machines simple inertia and shaft governors (see Figs. 330–334) are much used and serve their purpose sufficiently well.

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[1] See the investigation of M. Baraz on two of the older types of Deutz pendulum governors in Z. d. V. D. I., 1894, p. 1007.

Details of Some Special Governor Constructions.

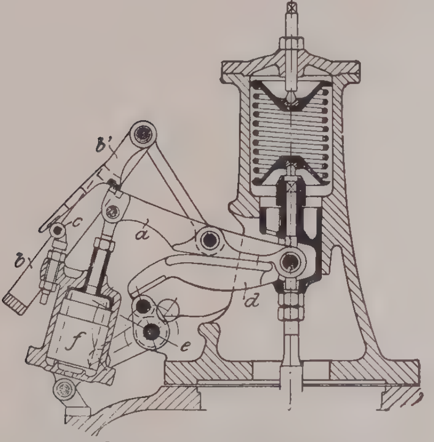
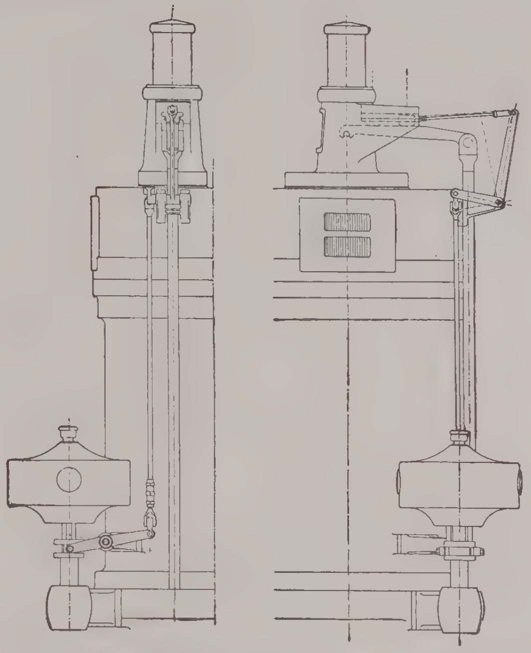


Fig. 326.—Governor Details for Large Engine, Maschinenbau-Ges., Nürnberg.

(The gas valve, fastened to the wiper cam or roller lever *a*, is lifted through latch *b'* by the eccentric rod *b*, until the roller *c* breaks the connection between *b'* and *a*. The valve then closes under the action of the spring, the shock being absorbed by dash-pot *f*. Wiper cam *d* is supported by the bell crank *e*, whose position is changed by the governor, thus adapting the valve lift to suit the load.)



Figs. 327 and 328.—Governing Details for a Güldner Engine.

(The effective lift of the inlet valve is adjusted to the load by shifting the fulcrum about which the inlet valve lever turns.)

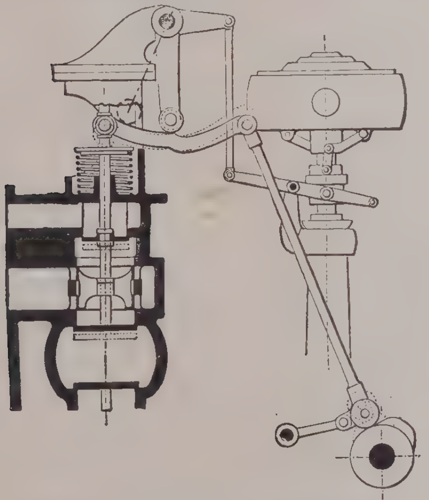


Fig. 329.—Governor Construction, Gasmotoren-Fabrik-Deutz.

(The governor shifts the position of the fulcrum about which the valve lever turns, thus suitably changing the lift of the gas or air valve or of the mixture inlet valve, as the case may be.)

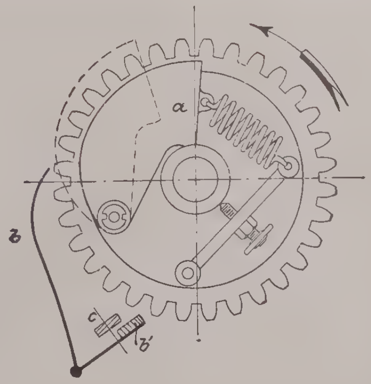


Fig. 330.—Körting Bros.

(Weight *a* pivoted to the larger gear of the two-to-one gearing as shown at n_{max} displaces the bell-crank lever *b–b'* so far that the block *b'* engages and holds the exhaust valve lever *c*.)

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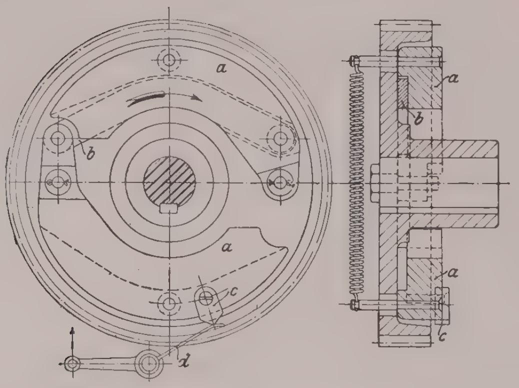
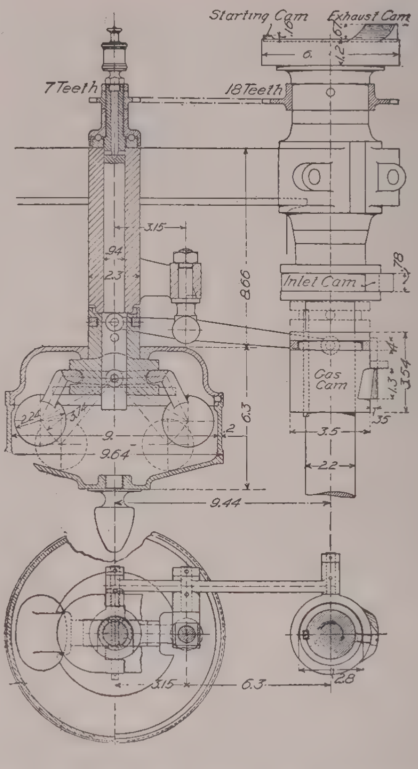
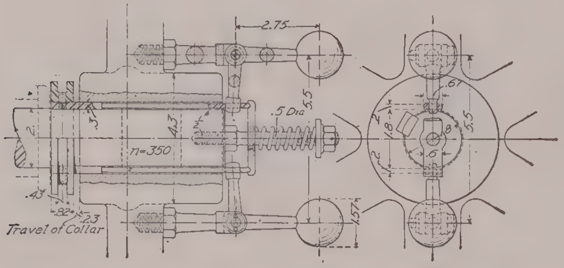
[1] Translator's Note. This entire discussion does not generally apply to American practice, where there are no firms making the building of governors a specialty.



FIGS. 331 and 332.—Bánki-Ganz & Co.

Weights a, in the larger gear, are connected by a link b. At $n_{\max}$ the projection c forces the lever d out far enough to cause the latter to engage and hold the exhaust valve lever.





FIGS. 333 and 334.—Shaft Governor, 6 H.P. Güldner Two-cycle Engine.

FIGS. 335 and 336.—Suspended Centrifugal Governor, 12 H.P. Loutzki Engine.

The *coefficient of governor regulation* δ_r, is often based on the difference in the number of revolutions per minute of the engine for the lowest, mean, and highest position of the governor sleeve. This agrees fundamentally with the general expression

$$\delta_r = \frac{n_{\max} - n_{\min}}{n} = \frac{n_{\max} - n_{\min}}{\dfrac{n_{\max} + n_{\min}}{2}} = 2\left(\frac{n_{\max} - n_{\min}}{n_{\max} + n_{\min}}\right). \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

Governors bought in the market usually have a value of δ_r varying from 2 to 4%. Although a small value of δ_r is in itself of advantage for close governing, governors with small coefficients are apt to act not only under changes of load but also to react with the variation of the turning effort within one cycle. This leads to a restless governor play familiarly known as "hunting." To avoid any possibility of its occurrence, it is best to employ governors whose values of δ_r may be changed within rather wide limits, say from 3 to 6%.

As a criterion of the stability of centrifugal governors the so-called centrifugal curve[1] has lately been extensively used, and some designers maintain that the degree of stability of a governor may be reduced to the astatic or isochronous condition when this C-curve is a straight line. This, however, is not correct, since in the derivation of this C-curve it has been assumed that the effective masses of rotation are the same for all positions of the governor sleeve, a condition which does not hold true for any of the well-known governors. On the contrary, the value for the effective mass of rotation changes with every different sleeve position when the influence of the arms, levers, etc., upon the condition of equilibrium at every different position is correctly taken into account.

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[1] TRANSLATOR'S NOTE. By this is meant a curve showing the relation between the total centrifugal force of all the rotating masses involved and the distance of the center of gravity of the equivalent mass from the axis of rotation. A very clear exposition of the C-curves and their use, and of governors in general is given by M. Tolle, in Die Regelung der Kraftsmaschinen.

Hence not the C-curve itself but a " reduced C-curve," taking into account these variations, should be used as a basis of comparison; and it is only possible to reduce the degree of stability of a governor to the astatic condition when the latter curve is a straight line.[1]

At any one position of the governor sleeve the total centrifugal force C must increase or decrease by a certain amount ΔC before the governor can overcome the resistance P at the sleeve necessary to change the position of the latter. The ratio of ΔC to C is evidently a measure of the sensitiveness or of the sluggishness of the given governor, as one chooses to take it. The *coefficient of sensitiveness*, if we may so call this ratio, then is

$$\varepsilon=\frac{\Delta C}{C}=\frac{(n+\Delta n)-(n-\Delta n)}{n}\sim\frac{(n+\Delta n)^2-n^2}{n^2}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

Evidently the smaller the internal friction of the governor the smaller the coefficient of sensitiveness and the more sensitive the governor. In general, ε should be as small as possible, but in any case it should be somewhat greater than the coefficient of fly-wheel regulation δ. If this rule, which is of special importance in 4-cycle machines, is disregarded, the governor will hunt constantly, a state of affairs that can only be remedied, as a matter of last resort, by oil dash-pots or other similar means. The best types of centrifugal governors bought in the market have a value of ε varying from 2.0 to .5%. If now a governor of this type is applied to an ordinary stock engine, in which the value of δ is generally from 2.5 to 4%, the inevitable result is a reaction of the governor within each cycle, leading to ceaseless hunting, just as in the case of too small a value of δ_r. If there is uncertainty regarding the value of δ, it is best to use governors which allow of some adjustment regarding sensitiveness.

Concerning hit-and-miss regulation, on the other hand, a centrifugal governor can hardly be made too astatic or isochronous for such service because the quicker such a governor changes from one extreme position to the other, the smaller will be the lag in the action of the governor and the less frequent will be the disturbances in the operation of the governor during the transition periods. The limit to the sensitiveness is, however, in this case also set by the fact that unnecessary hunting should be avoided.

The greatest power of regulation, that is, the ability of the governor to most quickly attain the state of equilibrium corresponding to a new load on the engine, is possessed by that governor in which the equivalent sleeve lift h_r is smallest in comparison to the actual lift. The value of the equivalent lift is

$$h_r=\frac{\text{Sum of all weights}\times\text{square of distances moved}}{\text{Total energy}}$$

To attain maximum power of regulation therefore it is necessary to have all masses to be accelerated and their range of movement as small as possible, that is, speed of rotation and distance of the masses from the axis of rotation should be as large as possible and the path of the masses should be a straight line perpendicular to this axis. In the governors of usual design h_r varies from $\frac{1}{15}h$ (Jahn's spring governor *C*) to h, where h is actual range of sleeve movement.

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[1] See the investigations of Jahns in the governor catalogue of Wilh. Rivoir, Offenbach M.

Tables 24 and 25 are intended to give some data which may serve as an aid in the selection of commercial centrifugal governors for a given service. The figures given for the force at the sleeve possessed by the different types are based upon the assumption that the speed changes 2%, that is, $\Delta n = .02\,n$. From this the coefficient of sensitiveness is

$$\varepsilon = \frac{2\Delta n}{n} = \frac{2 \times .02n}{n} = .04.$$

The energy E that must be possessed by a governor to furnish the given force P at the sleeve may be expressed by

$$P = \varepsilon E,$$

so that in this case

$$E = \frac{P}{\varepsilon} = \frac{P}{.04} = 25\,P.$$

The data furnished generally neglects the effect of governor friction upon ε or E, that is, ε is usually given too small. In more accurate terms, the available force P, and consequently the energy E are expressed by

$$P = E(\varepsilon - \varepsilon_r), \quad . \quad . \quad . \quad . \quad (3) \qquad \text{and} \qquad E = \frac{P}{\varepsilon - \varepsilon_r}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

If the shifting of the governing mechanism concerned requires an expenditure of work equivalent to say, A in.-lbs., the required power of a governor having a sleeve lift equal to h inches may then be found directly from

$$hE = \frac{A}{\varepsilon - \varepsilon_r} \quad \text{or} \quad E = \frac{A}{(\varepsilon - \varepsilon_r)h} \text{ lbs.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

In good spring governors, $\varepsilon_r = .005$ to $.02\,\varepsilon$, although in specially bad cases the value may go much higher. If, in general, the efficiency of a governor is expressed by

$$\eta = 1 - \frac{\varepsilon_r}{\varepsilon}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

we may from the above also write

$$= \frac{P}{\eta\varepsilon} = \frac{A}{\eta\varepsilon h}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5a)$$

According to investigations carried on by Jahns and others, for a value of $\varepsilon = .04$, the efficiencies of twelve governor types examined varied between 95 and 50%, showing an extraordinarily wide range.

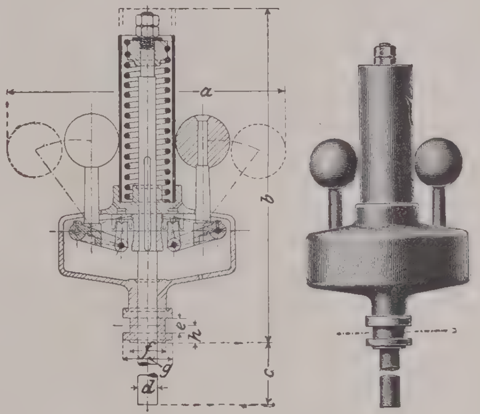
The data given for the force P in Tables 24 and 25 are based upon the normal number of turns stated. If this in any given case is different from the tabular value, for instance, on account of operating the governor from the half-time shaft, it should be remembered that P, as well as A and E, vary in the ratio of the square of n.

Example. Suppose that the work to be done in operating a given governor mechanism, including the reach rod from governor sleeve to the machine part to be moved, is found to be 56.6 in.-lbs. The governor is operated from a half-time shaft making 200 r.p.m. From Table 24 it will be seen that a Beyer governor No. 14 may be used for this work, since the power of this model is, owing to the reduction in speed from the normal 220 r.p.m. reduced to

$$A' = A\,\frac{200^2}{220^2} = 81.8\,\frac{200^2}{220^2} = 67.7 \text{ in.-lbs.}$$

It is assumed in this case that the governor friction is small, that is, that the governor efficiency η is correspondingly high. Since it is usually possible to only approximately determine the resistance to be overcome, and since the resistance often varies considerably owing to changed conditions of operation, it is always advisable to choose a governor that gives considerable power over and above what seems actually required.

In certain applications of internal-combustion engines, especially where such engines are used to drive generators, it is often one of the requirements that the revolutions of the engines can be changed by hand (for instance, during the paralleling of alternating current generators or the charging of storage batteries, etc.). This requirement is generally met by the governor manufacturers furnishing some means for increasing the resistances encountered, as spring balances, sliding poises, etc. In such manipulation of the governor the effect of the changed number of turns upon the coefficient δ of fly-wheel regulation should not be overlooked.



Figs. 337 and 338.—Centrifugal Governor, Franz Beyer & Co., Erfurt.

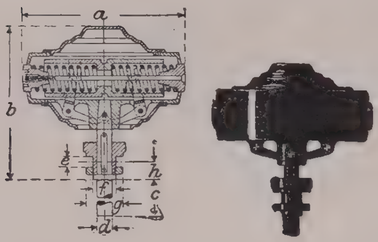
The models mostly used in gas engines are Nos. 10 to 15. In each of these, both n and δ may be changed within comparatively wide limits; the former by changing the position of the nut on the spindle, the latter by increasing or decreasing the number of effective coils of the spring. For this purpose the spring plate at the top is so constructed that it may be turned into or out of the spring. The number of coils and the value of δ_r bear the following relation:

$\delta_r =$	2	4	6	8	10%
No. of coils required =	13.5	13	12.5	12	11.5

This governor may also be used in a horizontal or in an inverted position.

Table 24

Beyer's spring governor.......Size No.	10	11	12	13	14	15	16	17
Rev. per minute....................n	300	280	260	240	220	200	180	160
Mean force P at sleeve ....lbs.	3.75	7.70	13.2	20.9	29.8	42.0	59.5	83.7
Lift of sleeve, h..................ins.	1.17	1.57	1.97	2.36	2.75	3.14	3.73	4.53
Power $A = Ph$..................in.-lbs.	6.55	12.10	26.00	49.20	81.8	132	222	380
Mean energy $E \sim \frac{P}{.04}$..............lbs.	88	185	338	512	750	1100	1500	2100
Main dimensions, inches, see Fig. 337.								
Max. width..........................a	12.25	15.50	19.25	22.40	25.70	30.00	34.90	40.70
Max. height.........................b	16.00	20.30	23.60	26.30	30.70	35.40	40.70	47.00
Free length of spindle..............c	15.70	17.70	21.60	25.50	30.50	35.40	41.40	47.30
Dia. of spindle.....................d	.98	1.18	1.37	1.57	1.77	1.97	2.36	2.76
Width of groove in sleeve...........e	.87	.95	1.02	1.18	1.37	1.57	1.77	2.36
Groove dia. of sleeve...............f	1.57	1.97	2.36	2.55	2.95	3.34	3.74	4.54
Ext. dia. of sleeve.................g	2.54	2.95	3.34	3.74	4.33	5.12	5.72	6.70
Distance from center of sleeve to lower face..........................h	.98	1.18	1.18	1.37	1.57	1.97	2.36	2.95
Weight, without packing..........lbs.	44	66	110	176	253	363	506	660



FIGS. 339 and 340.—Centrifugal Governor, Hermann Hartung Nachf., Düsseldorf.

Only the smaller models, up to say No. 98, are commonly used for internal-combustion engines. The number of revolutions *n* of the engine can not be directly controlled, but it may be increased up to 15% by the interposition of a spring balance between sleeve lever and governor support. This governor is also built with all levers completely enclosed.

TABLE 25

Hartung's spring governor. Size, No.		91	92	93	94	95	96	97	98	99	100
Rev. per minute . . . . *n*		340	310	240	240	210	200	190	180	165	160
Mean force *P* . . . . lbs.		5.05	7.05	9.25	12.50	15.2	19.6	22.9	25.3	33.2	50.0
Lift of sleeve . . . . ins.		.79	.98	1.18	1.18	1.57	1.97	2.36	2.76	3.15	3.54
Power . . . . in. lbs		4.0	6.90	10.9	14.8	23.8	38.4	54.2	70.0	105	152
Mean energy . . . . lbs.		126	176	231	312	379	489	572	638	832	1240
Main dimensions in inches, see Fig. 339	*a*	10.45	12.20	13.8	15.0	16.5	18.1	19.7	21.7	26.0	30.8
	b	10.90	11.80	14.2	15.3	16.5	17.3	19.1	20.2	22.3	24.3
	c	19.7	21.7	23.7	23.7	25.6	27.6	31.6	31.6	35.5	39.4
	d	1.02	1.11	1.38	1.38	1.57	1.77	1.77	1.97	2.17	2.37
	e	.79	.88	1.02	1.26	1.26	1.26	1.34	1.34	1.42	1.50
	f	1.82	1.89	1.97	2.17	2.37	2.57	2.97	3.15	3.35	3.55
	g	3.00	3.08	3.35	3.75	4.25	4.35	4.57	4.73	5.35	5.90
	h	1.57	1.73	1.89	2.01	2.13	2.21	2.25	2.29	2.41	2.45
Weight, without packing . . . . . lbs.		55	75	115	134	185	229	286	370	505	700

XI. Ignition Apparatus

In present day internal-combustion engines ignition is produced in one of the following ways:

(*a*) By means of incandescent or hot tubes, and in some oil engines also through certain highly heated places in the walls of the vaporizer.

(*b*) By means of electric spark.

(*c*) By heat of compression in conjunction with the heat returned by the cylinder walls. This method does not require any special ignition apparatus, and does not therefore have to be considered here.

1. Hot-tube Igniters. Hot-tube igniters are usually made from small tubes having an internal diameter varying from $\frac{3}{16}$ to $\frac{3}{8}$ in. and a length of from 2 to $3\frac{1}{2}$ ins. These hot tubes, which may be made of porcelain, platinum, or wrought iron, are to-day in use only on small and medium sized gas and oil engines, but on account of their low cost, simplicity, and reliability are superior to electric ignition for that service. For gasoline engines the open flame usually employed to heat the tube may under certain circumstances be dangerous on account of the nature of the fuel used to operate the engine, while in large gas engines the hot zone of the tube is usually too far from the center of the charge and the hot surface is too small in comparison with the volume of gas to be ignited to give satisfactory service. Hence in either case electric ignition is better.

Designs of Hot-tube Igniters:

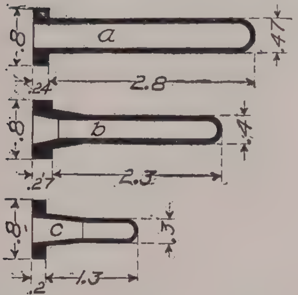


Fig. 341.—Commercial Forms of Porcelain Hot Tube Igniters. Carl Richter, Charlottenburg.

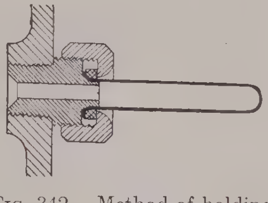
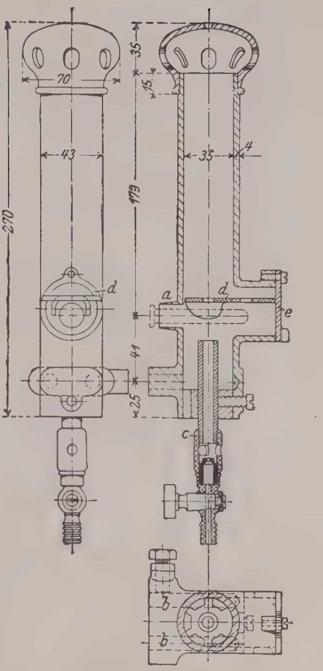


Fig. 342.—Method of holding Platinum Hot Tube.



Figs. 346–348.—Burner with Chimney, for the Hot Tube on a Grusonwerk Engine.

(Boss *a* seats in the cylinder head, see Figs. 130–133, *b* rods supporting chimney, *c* Bunsen burner, *d* flame spreader, *e* slide for observing adjustment of flame. The burner may be adjusted in the vertical, the entire chimney in the horizontal plane.)

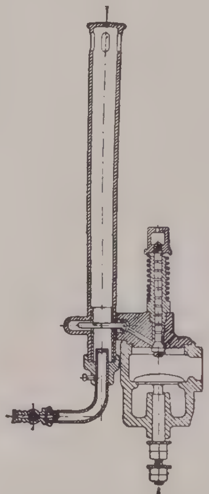
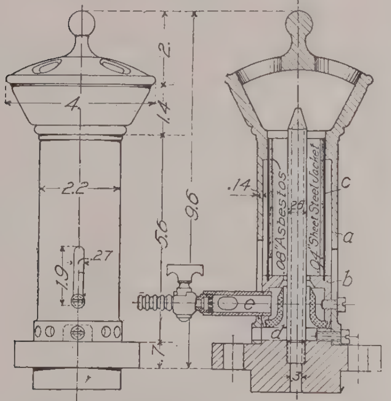


Fig. 345.—Hot Tube with Timing Valve, used on one of the older Körting Engines.



Figs. 343 and 344.—Iron Hot Tube with Chimney and Burner, Loutzki Engine.

(Chimney *a* together with lining *c* may be turned on the support *b*. Auxiliary air is supplied through *d*. *e* is an ordinary Bunsen burner.)

The hot tube should be so connected into the combustion chamber that the tube or passage can, during the suction stroke, be cleared as completely as possible from any burned gas, that only a good rich mixture is present at the mouth of the tube at the end of compression, and that the flame may spread freely in all directions and by the shortest possible paths. It is also essential that the inner opening of the tube be protected against water and lubricating oil which may stop up the narrow opening and in the case of porcelain tubes may lead to breakage.

As already stated, hot tubes are to-day used only for the smaller machines, and the usual construction is to use them without valves, that is, in constant communication with the cylinder (open hot tube). The time of ignition in such construction in a way adjusts itself automatically, because ignition of the entire charge can only take place when the velocity of the mixture entering the tube is less than the velocity of flame propagation of the ignition flame.[1] The latter is formed when the fresh mixture entering the tube has compressed the burned gases remaining in the tube into the outer end and up to the hot zone. In the dead center-position of the piston the velocity of gas entering the tube has of course reached its minimum, the compression in the tube is the maximum, while the mixture has reached furthest into the tube. Consequently ignition of the charge will always take place near the dead-center piston position, provided of course that diameter and length of tube, position of the hot zone and the gas content of the charge have been properly adjusted.

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[1] See Z. d. V. D. I., 1893, pp. 1425, 1615.

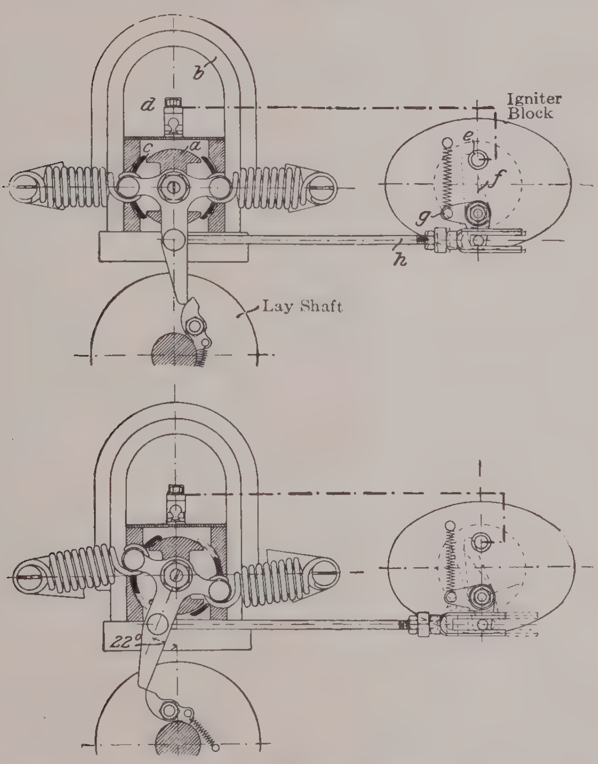
Hot-tube igniters with timing valve have almost entirely gone out of use. They do not offer, as shown, any marked advantage, increase the cost of the igniter gear, and on account of the fact that the timing valve, which is exposed to the hot ignition flame, is easily thrown out of order, their use simply impairs the reliability of operation of the machine. It is also possible in the open hot tube to adjust the time of ignition within comparatively wide limits. To avoid pre-ignition, for instance, it is merely necessary to shift the hot zone a little toward the outer closed end of the tube, or to slightly contract the inner opening. On the other hand, the timing valve is of service in that it prevents the dangerous pre-ignitions sometimes incident to starting. If these are also to be avoided with the simple open tube, it is always necessary to start with decreased compression. This precaution of course is especially important in engines which compress highly during normal operation.

The flame usually used to heat the tube is simply a Bunsen burner. The latter should be adjustable in two directions, so that the hot zone may be produced anywhere along the tube, and so that the hot core of the flame may be brought further away or closer to the tube (see Figs. 346–348). When properly adjusted the burner should show a blue flame and the fuel consumption per hour should be from 5 to 7 cu.ft. of illuminating gas, from .20 to .30 lbs. of gasoline, or from .45 to .60 lbs. of kerosene. In the case of illuminating-gas engines, the pipe line supplying the ignition flame should be taken off the main supply pipe ahead of the gas bag or other pressure regulator (see Plate VIII in Part III).

2. Electric Igniters. What has been said with reference to the position of hot-tube igniters in the combustion space, applies with equal force to electric igniters. Although the electric spark has a higher temperature than the hot tube, the effective surface is much smaller, and the proper position of the igniter is hence of even greater importance. Wherever the shape or size of the combustion chamber tends to interfere with the uniformly rapid ignition of the charge, it is better to use two or even more igniters properly distributed. This also tends to increase the reliability of operation.

Depending upon the source of current and the igniter mechanism, either a single spark or a series of sparks is employed. Stationary engines usually produce their own ignition current by means of simple electric generators and also produce the spark by operating suitable make-and-break mechanism. For automobile engines jump-spark ignition, used by Lenoir in 1865, has of late years come into extended use. Since the voltage furnished by the ordinary cell is altogether too low, it becomes necessary to step it up by means of induction. While automatic or mechanical interrupters (called tremblers) are sending a series of current impulses through the primary winding of an induction coil (spark coil), a series of sparks will bridge the gap between the two electrodes of the "spark plug" in the cylinder, which plug is connected to the secondary winding of the coil. This jump-spark, while not as strong as the make-and-break spark, is satisfactory for the ignition of the rich mixtures used. The life of an ignition battery consisting of four cells is approximately about 400 operating hours, which is a comparatively short time. In place of these cells special storage batteries are now often employed for automobile-engine ignition. With a mean voltage of from 4–5 volts in the primary circuit, jump-spark coils require from .03 to .06 amperes for their operation, the engine making in the neighborhood of 1500 r.p.m. For stationary engines, jump-spark ignition is considered unreliable and uneconomical, and consequently has now been almost entirely displaced by magneto ignition. In the traction engines of Lehmbeck an indirect magneto system of ignition has lately been used in which the current from the Bosch magneto is sent through a spark coil, while the rest of the gear is like that of ordinary jump-spark ignition. This idea is not patentable.[1]

Designs of Electric-magnetic Ignition Apparatus:



Figs. 349 and 350.—Bosch Ignition Apparatus, Stationary Armature.

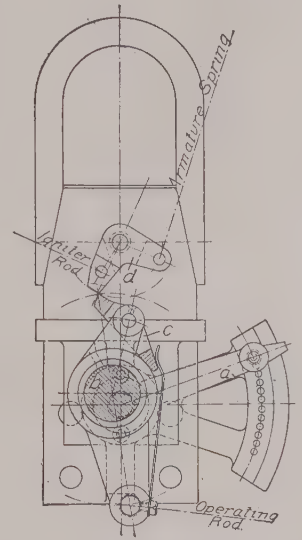
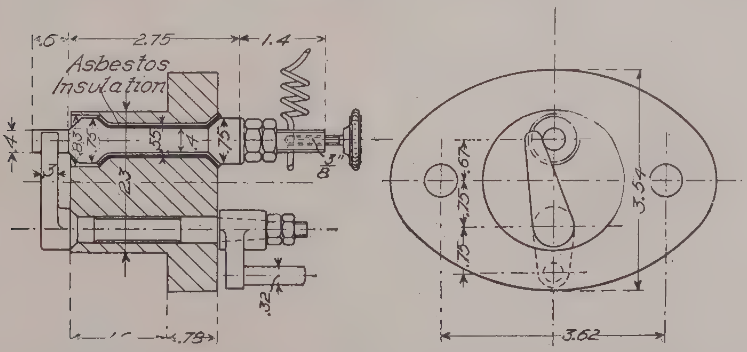


Fig. 351.—Igniter Gear, Maschinenbau-Ges. Nürnberg.

(Eccentric shaft *b* is shifted about its axis by adjusting lever *a*, causing the latch *c* to snap off the armature lever at the desired instant. The operating rod is actuated by means of a crank with a throw of 1″, from the end of the lay shaft.)



Figs. 352 and 353.—Standard Bosch Ignition Plug for the Smaller Engines.

(The stationary electrode may also be insulated by encasing it in porcelain, soapstone or enamel.)

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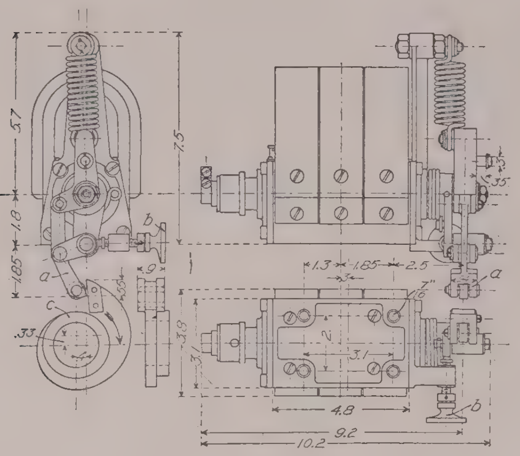
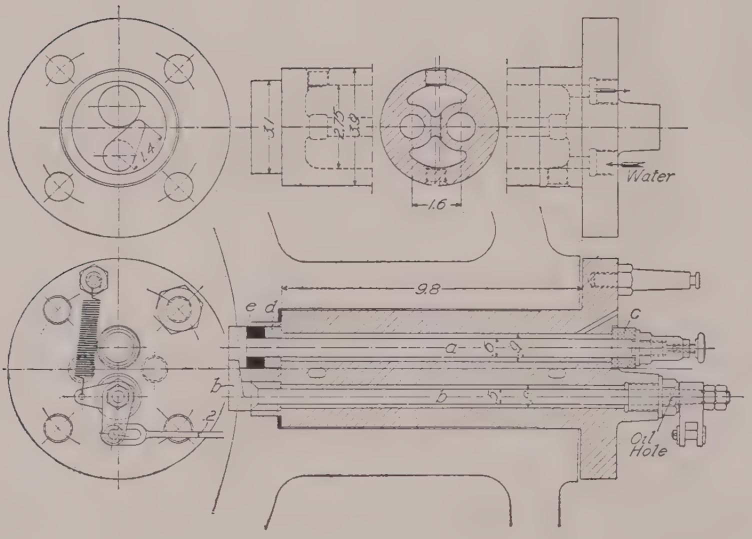
[1] Translator's Note. In this country a number of different jump-spark ignition systems are used. See any special book treating on the subject of ignition.



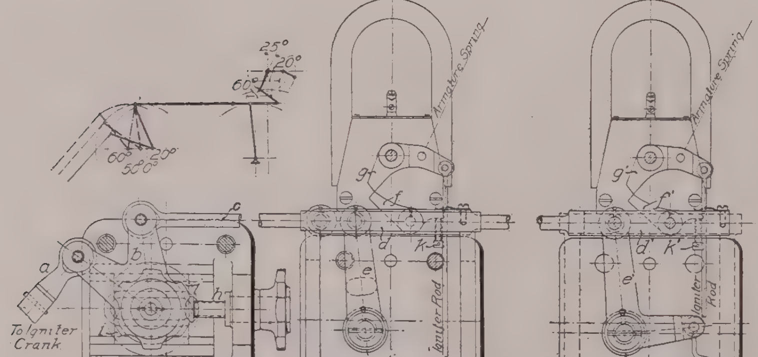
Figs. 354 and 355.—Ignition Apparatus with Adjusting Gear. Unterberg & Co., Stuttgart.

(Bell-crank *a*, instead of swinging about the center of the armature as usual, moves up and down following the cam *c*. By means of adjusting screw *b*, the distance between *a* and the vertical line through the armature center may be changed, thus varying the point of ignition.)



Figs. 356–360.—Water-cooled Ignition Block for a Large Blast Furnace Gas Engine, Maschinenbau-Ges., Nürnberg.

(*a*, enameled stationary electrode made of steel; *b*, movable electrode made of Durana metal; *c*, hard rubber insulation; *d*, mica insulation; *e*, porcelain insulation.)



Figs. 361 and 362.—Igniter Gear used by Maschinenbau-Gesell., Nürnberg for Double Ignition.

(When the rod *a*, through the bell-crank *b*, pulls the linkage *cdd'* and the bell-cranks *e* and *e'* to the left, the dogs *f* and *f'* will turn *g* and *g'* through 25°, after which *f* and *f'* snap off. By means of spindle *h* the fulcrum spindle *i* of the bell-crank *b* may be shifted, thus adjusting the point of ignition between −20° and +60° of crank-angle. Each igniter, however, may be adjusted independently by changing the positions of *f* and *f'* by means of the screws *k* and *k'*.

For magneto ignition the various types of apparatus built by Robert Bosch in Stuttgart have practically become the standard. The latest model, originally built for high-speed engines, has also found extensive application to stationary engines.

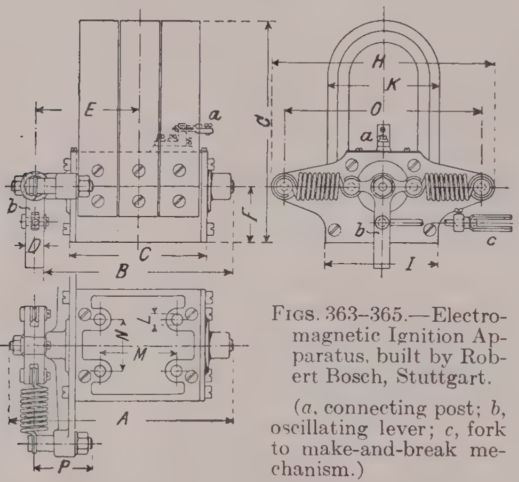
The Siemens I-armature *a* (Figs. 349, 350), which in the older constructions had to swing through an angle of approximately 50° for every current impulse produced, in the later type remains stationary between the poles of the built-up horseshoe magnet *b*. The lines of force are deflected by a split sleeve *c* made of soft iron, which slips over the armature *a*. For this purpose it is necessary just before the spark is desired to turn the sleeve about its own axis through an angle of from 20 to 25% from its normal position, which in stationary engines is usually done by a cam on the lay shaft. In high-speed engines the sleeve is operated positively either by crank or eccentric. The turning of the sleeve puts the two helical springs shown in strong tension so that at the moment the cam releases the sleeve the latter is brought back to its original position with great rapidity. During this movement a strong current impulse is induced in the winding of the armature *a*. One end of this winding is grounded on the frame of the magneto itself, while the other is connected to the contact *d*. The latter in turn is connected with the insulated electrode *e* of the make-and-break plug, as shown. The inner arm of the lever *g* is held in contact with *e* by means of a spring. A forked rod *h* is connected on one end to the vertical arm of the sleeve lever, as shown, while the fork rests on the pin of the outside bell crank *g*. At the instant the cam on the lay shaft releases the sleeve lever, the inner end of the fork strikes the pin on which it slides, and the inside arm *f* is suddenly brought out of contact with the pin *e*, which causes a spark to form between the two. On account of inertia effects, the time of sparking lags a little behind the instant of snapping off of the sleeve lever; to cut this lag down to the lowest possible amount all moving parts should be made as light as permissible.

The length of the make-and-break spark depends upon the rapidity with which contact is broken; for that reason the inside lever should be made as long as possible, a good length is twice that of the outside arm *g*. If it is not possible to use this proportion, make the vertical sleeve lever (*b* in Fig. 364) correspondingly longer.

The interruption of the current should be made to take place at the instant that the sleeve is most strongly affected by the magnetism of the poles, that is, where the sleeve offers the greatest resistance to turning. (This position may easily be found by revolving the sleeve by hand.) The fork in its normal position should not bind the pin upon which it slides, to make sure that *f* is always in contact with *e*. The striking of the pin by the fork should only be done by the sleeve lever when the latter on its return to the point of rest momentarily over-travels this position. The greater the frequency of the spark required in a given time, the smaller should the deflection of the sleeve be made; to counteract this the electromagnetic power of the apparatus must of course be correspondingly increased, a condition which should not be forgotten while ordering. If for some reason or other it should at any time become necessary to remove the armature, a piece of iron of approximately the same size should be put in its place to keep the magnetic circuit closed.

The stationary electrode *e* requires high grade insulation (covering with asbestos, porcelain, or soapstone), because the ignition current possesses high voltage. Care should be taken to see that no short circuits can be formed through incrustations of burned oil, soot, etc. The spindle of the movable electrode is often made with a conical seat at the inner end, which serves as a packing surface (see Fig. 352). Where the compression is high, however, a flat seat is better, especially if the collar rests against a steel sleeve, because this construction minimizes the danger of the very troublesome sticking fast of the electrode, and the grinding in of the conical seat (see Fig. 360). Thorough cooling of the igniter block is of importance. The stationary electrode and the inside hammer lever should always be made quite large, since these parts if too light are apt to become red hot, which under high compression may lead to pre-ignition.

Since the ignition or explosion of the charge should be completed as far as possible at the dead-center position, it becomes necessary to cause the spark to occur before this position is reached; and, in general, the leaner the mixture and the higher the piston speed, the more should the spark be advanced. At starting the spark should be so far retarded that dangerous pre-ignitions cannot take place. While this spark adjustment may in general be easily carried out by very simple means, large gas engines should be equipped with apparatus which admits of fine adjustment in order to be able to control the combustion with accuracy at any time during operation as well as during testing out. In the Unterberg magneto (Figs. 354, and 355) such spark control apparatus is directly combined with the ignition apparatus, which for the smaller gas engines is probably a desirable construction. In large gas engines the spark adjustment had better be made by changing the position of some parts of the ignition gear. Examples of this are shown on pages 253 and 254. Automatic adjusting of the spark through the engine governor is beset with great practical difficulties, and for that reason has so far not found successful application.



Figs. 363–365.—Electromagnetic Ignition Apparatus, built by Robert Bosch, Stuttgart.

(*a*, connecting post; *b*, oscillating lever; *c*, fork to make-and-break mechanism.)

The principal dimensions in inches of the new Bosch magneto with exposed springs, for r.p.m. ≦ 175, may be taken from Figs. 363–365 and Table 26.

Table 26

Dimensions in Inches.

Type.	*A*	*B*	*C*	*D*	*E*	*F*	*G*	*H*	*I*	*K*	*L*	*M*	*N*	*O*	*P*
C.	7.10	5.67	3.78	.79	3.25	1.77	6.02	7.47	3.15	3.19	$\frac{3}{8}$	1.97	1.97	6.70	1.77
M_3.	8.66	7.20	5.28	.79	4.01	1.77	8.45	8.45	4.25	4.33	$\frac{1}{2}$	3.15	1.97	7.48	2.17
M_3.	. . . . .	8.85	6.90	.79	4.85	1.77	8.45	8.45	4.25	4.72	$\frac{1}{2}$	3.15 or 4 75	1.97	7.48	2.17

These magnetos are also built with a single vertical inclosed spring. The construction shown with two springs overcomes the one-sided bearing pressure in the sleeve support and insures better operation.

XII. Pedestals and Foundations

Internal-combustion engines require foundations of somewhat better grade than steam engines of equal power, both because the working stresses in the former are considerably higher and because these stresses vary with greater rapidity. Further, the fly-wheel weight in gas engines is usually greater and the reciprocating parts are not cushioned on passing the dead center as in steam engines. All of these things combine to make the service required of the foundation more severe.

Small gas engines, especially vertical machines, may up to 15 H.P. be set upon the floor without stone foundation, provided the weight of the engine and the vibrations are distributed over a sufficient number of floor beams or over a sufficiently large area by means of sub-bases or pedestals (Fig. 366). To deaden any resonant vibrations, the hollow spaces in the sub-base are best packed with sand, and a sheet of asbestos, hard felt or cork, may be placed between the pedestal and the engine-bed proper. It goes without saying that larger machines which are held down in this way require more thorough balancing of moving parts than is ordinarily practiced in stationary engines.

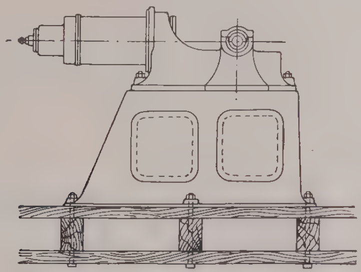


Fig. 366.—Horizontal Engine with Pedestal.

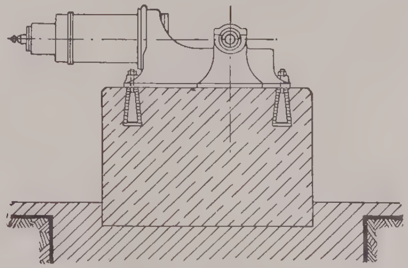
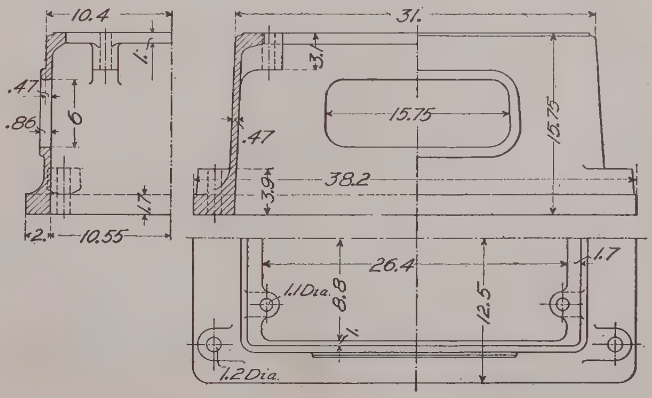


Fig. 367.—Horizontal Engine with Stone or Concrete Foundation.

An engine in steady operation should whenever possible be furnished with a stone or concrete foundation (Fig. 367). Such a foundation is usually cheaper and serves the purpose better than any other. The height of the foundation above the floor depends upon the height desired for the crank shaft, and in engines which are furnished with a cast-iron pedestal the foundation usually has only a course or two above the floor. The height of the pedestal should be taken accordingly.

Designs of Engine Supports or Pedestals:

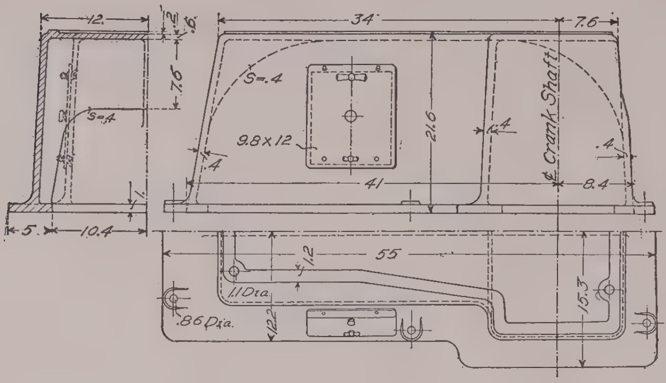


Figs. 368–370.—Pedestal for a 6 H.P. Vertical Kerosene Engine, Bánki, made by Ganz & Co., Budapest.

Engine $D=6.4''$

$S=10''$

(See also Figs. 45–47, p. 91.)



Figs. 371–373.—Pedestal for a 6–8 H.P. Horizontal Engine, Fr. Krupp-Grusonwerk, Magdeburg.

Engine $D=7''$
$S=12''$

Concerning the *construction of the foundation,* the following may be noted: In calculating the load to be carried by the soil, the weights of both engine and foundation must be considered; it must always be less per unit area than the compressive stress on the material of the foundation, and should not exceed

15	lbs. per sq.in.	for	soft clay and fine damp sand;
30	‘‘	‘‘	loam, medium hard clay and dry sand carrying clay;
45–60	‘‘	‘‘	hard clay and completely dry sand;
75–90	‘‘	‘‘	hard-bedded sand and gravel.

For first-class soil the mean allowable load may in general be taken at from 60–75 lbs. per sq.in.; in ordinary soil it is better not to exceed 45 lbs. per sq.in.

The material for foundations for small engines should of course be squared stone, where such can be had at reasonable cost, otherwise brick or concrete is used. Where stone is used, the mortar should contain as a binding material only hydraulic lime or cement, especially if the foundation is exposed to ground water. Hydraulic lime is cheaper and serves its purpose sufficiently well when the soil is constantly damp and there is plenty of time for the setting of the mortar (2–3 days). In all other cases hydraulic cement, Portland or Rosendale, is to be preferred. The mortar should preferably consist of good Portland cement and fairly coarse sand free from clay. If required the sand should be screened and washed, which may be done by immersing it in shallow wooden tanks and after a time drawing off the water. The proportion of cement to sand may vary from 1 to 3 (normal) to 1 to 2 for foundations carrying very heavy loads; for a finishing mortar for that part of the foundation above the floor a ratio of 1 to 1 may be used. The adhesiveness or strength of the mortar increases with the proportion of cement. Since natural stone does not bind as well as brick, it is better to use a rich mortar for the former (1 to 2 or 1 to 1).

The quantity of water used for the mortar varies somewhat, the best proportion seems to be from 25 to 30% of the weight of the cement used. Sand and cement are first thoroughly mixed, afterward the water is added. For low-grade foundations, or for the lower third, say, of any foundation, a leaner grade of cement mortar (1:4 or 1:6) or cement lime mortar (1 of cement, 5–6 of sand, 1 of slaked lime) may be used. One cubic yard of rubble masonry requires from .3 to .35 cu.yd. of mortar, while 1 cu.yd. of squared stone masonry requires from .08 to .15 cu.yd., depending upon the thickness of the joints.

Where brick is used only the hard burned should be employed; the ordinary building brick does not possess enough strength or durability. In the case of large foundations, however, the latter may be used for the inner layers or core, where it is protected from dampness and less heavily loaded. Each cubic yard of brick foundation requires about 500 brick and .3 cu.yd. of mortar, the total weight being about 3000 lbs.

The foundation should be carried down to solid soil and below the frost line. If the soil does not appear quite safe it should first be prepared by laying down a "gridiron" sub-foundation of timbers; while dangerous soil can only be taken care of by a sub-foundation of piles. If there is any uncertainty as to the quality of the soil, it is well to start the foundation with a larger surface than required and to step in the successive courses until the last one corresponds to the template. Instead of this the bottom of the hole may also be covered with a layer of small stone, broken brick, etc., to a thickness of from 1 to 3 feet. The repeated use of thin cement mortar soon binds this layer into one solid block, which should be larger by its own thickness on all sides than the foundation itself. Soil may be considered *safe* when it consists of rock or stony earth, sand or gravel (when dry), dry loam and clay (when the strata are at least 15 ft. thick); it may be considered *unsafe* when it consists of loam or clay when in thin layers or wet, quicksand, very wet sand, marl, alluvial deposits, etc.; and it is *dangerous* when it consists of turf or peat soil, swamp, filled soil of any kind, etc.

The "gridiron" sub-foundation above mentioned is made by placing timbers, 10″ or 12″ square, parallel to each other and about 4–6″ apart, level on the bottom of the pit. The intervening spaces and the clearance at the ends are then solidly filled up by tamping home concrete made up of 1 part by vol. of cement, 1 of lime, 6 of sand and broken stone. The timbers are then cross-connected by a layer of boards 3–4″ thick, spiked home. In pile foundations a similar grid is placed on piles which have been driven into the bottom. These piles are from 8″ to 16″ in diameter, and from 10′ to 20′ long. The distance between piles and rows of piles is from 25″ to 50″. The individual piles are connected by strong cross-beams and upon these are placed planks as above. The position of these should be at least from 12″ to 20″ below ground water. In both cases the foundation proper rests upon the planks. The best timber for the piles, beams, and planks is oak, pine, or hemlock, each of which may first be impregnated if desired. In every case the factory furnishes the foundation plans. When desired the buyer should also be given directions for setting and lining up, which should make mention of the following points:

After the engine is placed on the foundation and the foundation bolts are entered into the holes in the frame, the crank-shaft and the center line through the cylinder should next be accurately leveled up (paying due regard of course to any shafting that may have to be operated). The best way to do this is to use thin iron wedges which are driven under the engine frame. Care should be taken to see that the foundation bolts are long enough to furnish sufficient thread for nut and washer. After the engine is leveled the nuts are then lightly drawn up by hand.

The next operation is to surround the entire foundation with a collar by using thin boards, or by piling sand or loam along the edges, and to pour in very thin cement mortar until the entire foundation is covered up to the level of the engine frame. Care should be taken to see that the mortar covers every part uniformly, for which purpose a piece of thin band iron may be drawn back and forth under the frame a few times.

The mortar should next be allowed to set or harden completely, after which the foundation bolts may be fully drawn up. Care should be taken, however, to see that the operation does not warp the frame. For this purpose disconnect the connecting-rod from the crank, and keep turning the shaft in its bearings to see that it does not bind during the operation of drawing up the bolts. After this is completed the fly-wheel may be put on the shaft.

To facilitate the putting on of the wheel, construct an inclined plane of such a height that the bore of the wheel comes fair with the shaft, after which the wheel may be shoved into place.

See that the bore of the wheel, the wheel seat, and the key seat are free from dirt or paint and oil them thoroughly. In order to prevent the shaft from turning while the wheel is forced into place the former may be locked by placing a block of wood under the crank in the crank case.

The **dimensions of foundations** not only depend upon the type and size of engine, but also, and very materially, upon the kind of soil encountered. If the latter is of medium quality, still considered safe, an engine having a cylinder diameter of D inches, whether of the horizontal or vertical type, requires a depth of foundation below the floor of from 4 to 5 D. In all cases, however, the foundation should reach below the frost line (3 to 5 ft.).

The length and width of the foundation of course depend directly upon the size of frame or pedestal. The volume of the foundations for the different types of engines may be taken on the average as follows:

For horizontal engines, without outboard bearing, 14–18 N_n cu.ft.;
" " " with " " 21–25 N_n cu.ft.;
" vertical " without " " 7.7–8.8 N_n cu.ft.;
" " " with " " 9.8–10.5 N_n cu.ft.

where N_n = normal brake horse-power.

It is a matter of experience that horizontal vibrations are more severe on foundations and soil than vertical ones, hence for uncertain or unsafe soils, vertical engines are better suited than horizontal machines. See p. 316, covering costs of foundations.

Instead of having the foundation or anchor bolts pass down through the entire foundation, it is advisable for large engines to make these bolts shorter and thus to make them less yielding. For this purpose the lower end of the bolt is held by a cast-iron or steel plate which has been let into a recess in the foundation somewhere near the top. For standard sizes of anchor bolts, see page 90.

D. SPECIAL PARTS FOR GAS AND OIL ENGINES

I. Gas Engines

THE special parts of gas engines, which include mainly mixing arrangements and ignition gear, are intimately connected with cylinder head and valve construction, and have already been discussed, see pages 194 and 250. The auxiliary apparatus of gas-engine installations, such as gas producers and washers,[1] gas bags, gas meters, etc., will be taken up on pages 268 and following.

II. Oil Engines

The special apparatus to be considered under this heading also consists mainly of devices used to transform the fuel into a gas or vapor, to thoroughly mix the latter with air, and to ignite the mixture.

1. Carburetors. We distinguish vaporizing carburetors, in which the finely divided air is drawn through the volatile liquid fuel on the suction stroke of the engine; and spraying or atomizing carburetors in which the fuel is sprayed into the passing air in a fine stream. The latter method of forming the mixture may also be carried out without special apparatus in the inlet valve or the cylinder itself.

Vaporizing carburetors are to-day largely used for stationary machines, although they are large and heavy, increase the suction losses of the engine and are not capable of maintaining a constant mixture.[2] Since gasoline consists of a series of hydro-carbons, of various vaporizing qualities, it will be found that the mixture is rich in highly volatile vapors when the carburetor is newly filled, but the proportion of such vapor present decreases steadily as the level sinks, until at last there remain only hydro-carbons difficult to vaporize, and the result is a correspondingly lean mixture. The construction shown in Figs. 377–379 is supposed to avoid this difficulty.

Spraying carburetors are not used in stationary installations to the extent that their low cost, simplicity of construction, flexibility and ability to furnish a constant mixture would warrant.[3] For automobile engines, on the other hand, their use is universal and they have shown themselves reliable for speeds up to 2000 r.p.m. The construction of these carburetors has become a special manufacturing branch of the general field.[4]

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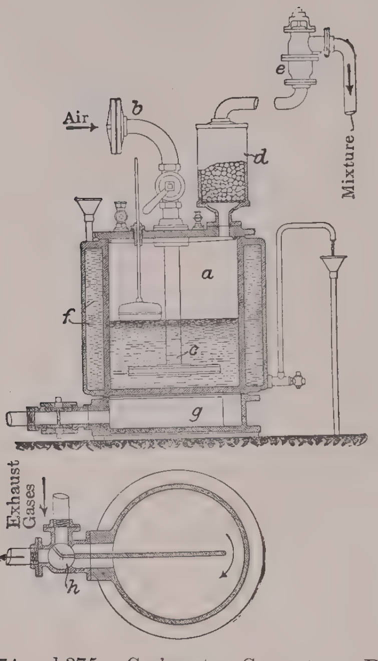
[1] The building of power gas producers is as yet largely a branch of the gas-engine industry, although there are manufacturers making the former a specialty. This separation of the two fields is justified when the building of producers is carried on by men acquainted with the general gas-engine field, but leads to unsatisfactory results when it is taken up by boiler works or similar shops merely as a side line. One of the undesirable features certainly consists in the increased difficulty of meeting all of the special operating conditions of the various types of engines. This quite often leads to trouble at the place of erection, costly both in time and money and in many cases causes the engine builder to abandon the use of producers of make other than his own for some time to come.

[2] TRANSLATOR'S NOTE. The first part of this statement hardly applies to American practice.

[3] See Note 1.

[4] See the pamphlet of Périssé, Les Carburateurs (Paris, Masson et Cie.), which gives an extensive treatment of the construction and tests of carburetors, especially those used in automobile practice.

Designs of Carburetors:



Figs. 374 and 375.—Carburetor, Gasmotoren-Fabrik Deutz.

(During the suction stroke the vacuum in *a* will cause air to enter through muffler *b* and screen *c*. Saturated with gasoline vapor the air next passes through *d* and the check-valve *e*, both inserted for safety, on its way to the engine. Here it is mixed with more air thus forming the normal mixture. To facilitate the formation of vapor in *a*, the warm cooling water from the engine is circulated through *f*. To the same end a part of the exhaust gases may be passed through the hollow bottom *g*, the adjustment being made at *h*. In winter it becomes necessary to first fill *f* with hot water, if the carbureter is located in a cold room.)

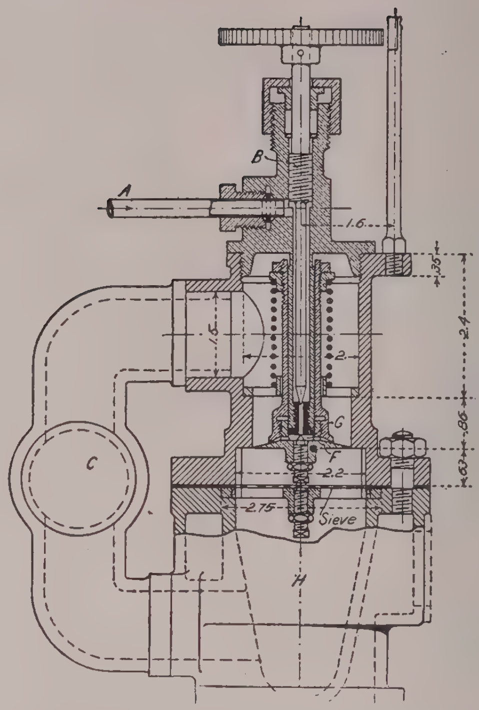
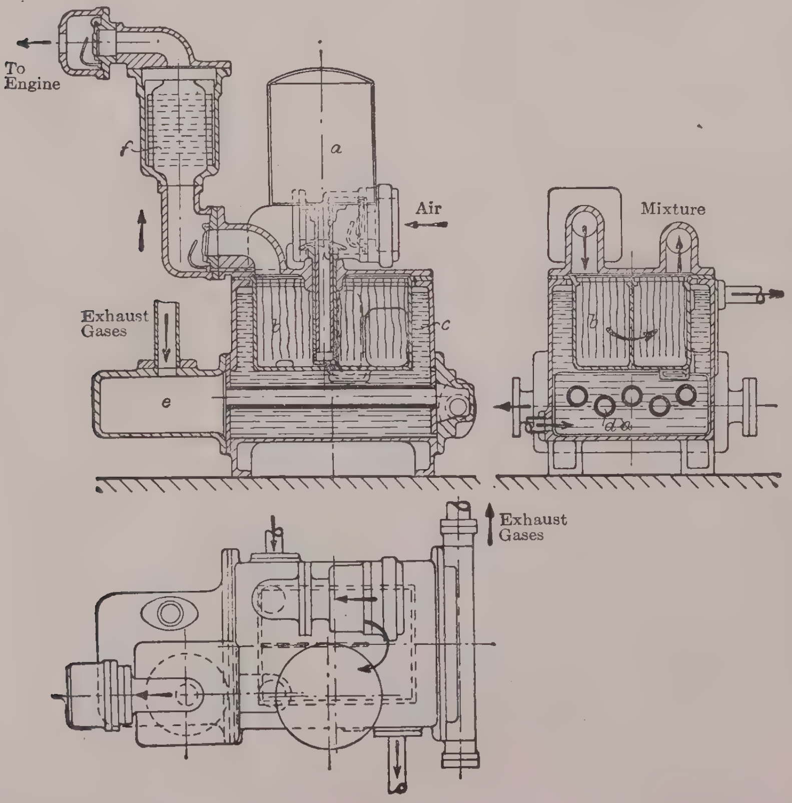


Fig. 376.—Spraying Carburetor, Martini & Co.

(The gasoline supply through *A* is regulated by means of the needle valve *B*. Air enters through *C*, and on the suction stroke of the engine the disk valve *F* moves downward. This motion causes the adjustable needle passing through *F* to free the lower end of the gasoline duct. The liquid then flows out through fine radial openings and is picked up and carried along by the stream of air. By adjusting *C* a part of the air may be allowed to pass directly into the passage *H*.)



Figs. 377–379.—Carburetor, Chr. Reichman, Munich.

(This carburetor is designed to furnish a mixture of more uniform composition than that which can be obtained in carburetors of design shown in Figs. 374 and 375, in which the more volatile parts of the liquid always distill off first. In this design the gasoline is fed drop by drop from the reservoir *a* through the central pipe and is absorbed by the suspended wicks *b*. From these it evaporates and mixes with the passing air. The wick chamber is surrounded by a water jacket *c*, which contains a nest of tubes *d* through which the exhaust gases coming through muffler *e* may be passed. Chamber *f* is filled with pebbles or with wire screens and acts as a safety device.)

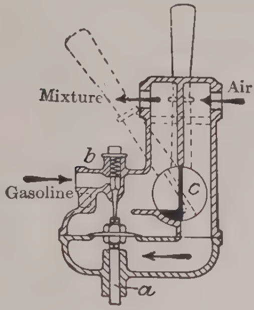


Fig. 380.—Spraying Carburetor, Sintz.

(Valve *a* operates needle valve *b*. By adjusting *c*, a part of the air may be led directly into the suction pipe.)

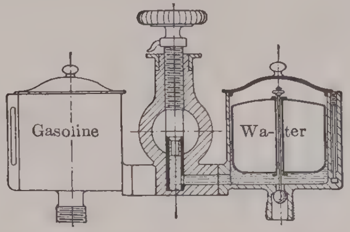


Fig. 381.—Double Carburetor, Bánki.

(The air drawn in by the engine passes through a pipe located between the two float chambers. Gasoline and water are supplied to this pipe by means of from two to four spray nozzles.)

2. Vaporizers and Atomizers. Both of these are intended to change the heavier liquid fuels, as kerosene and alcohol, to a state in which they will readily mix with air. In the former the oil is vaporized with the agency of heat, and the oil vapor is either immediately or subsequently mixed with air. Generally the oil is previously sprayed or atomized in order to spread it over a large surface. The well-known fact that liquids vaporize more easily in a partial vacuum has to the writer's knowledge never been practically utilized in oil engines.

If the oil is so completely atomized as to allow of the suspension of the particles in air, which, with suitable means may be easily accomplished, it is quite possible to form a satisfactory and uniform air-oil vapor mixture without the aid of heat. The oil thus mechanically carried along vaporizes only after reaching the cylinder during suction and compression. In order to get a clear insight into the action of vaporizers and atomizers, as well as oil engines themselves, it will be well to briefly consider the vaporization phenomena.

As shown in Part IV in greater detail, the crude oils are composed of a series of hydro-carbons having very different boiling points. The most volatile of them readily vaporize under temperatures ranging from 60° to 70° F., the greater proportion, however, require much higher temperatures for vaporization. Taking refined kerosene, for instance, usually more than 60% by volume will vaporize between 400° and 570° F., while solar oil loses 75% between 470° and 630° F. (see Fuels in Part IV). The degree of heat required to vaporize the heaviest constituents of the fuel oils is not far from a dark-red heat. We may assume that the temperature of the cylinder walls of an oil engine is somewhere between 140° and 340° F., the average probably in no case exceeds 300° F., but that is considerably below the condensation temperature of most of the constituents of the fuel oils. If therefore oil vapor, which has been produced in a special chamber or vessel sufficiently heated, is introduced into the engine cylinder, it is unavoidable that a part of the hydro-carbons of high boiling point in the vapor will, owing to its contact with the cold air simultaneously drawn in and its contact with the cylinder walls, returns to the liquid state and will be thrown down in the shape of drops. The reason for this occurrence of course lies in the fact that both the quantity and the specific heat of the oil vapor are too small to maintain the required mean vaporization temperature against the cooling effect of both air and walls, and a partial condensation of the oil vapor is consequently inevitable. The liquid thrown down burns only partially, owing to the fact that air can only reach its exterior surface, it gives off vapors during the exhaust stroke and finally forms incrustations on the inner walls.

"Superheating" the oil vapor does not have the desired result, first because the degree of superheating is limited on account of low temperature of ignition of the oil, and again because whatever degree can be employed has little effect upon the *mean* temperature for reasons already stated, and hence cannot in any marked degree prevent condensation. The scheme of superheating the oil vapor, patterned without good reason after steam-engine practice, has therefore nothing in its favor.

On the other hand, the opposite of this scheme that is to introduce the oil mechanically very finely divided (in shape of "fog") into the air and thus into the cylinder, promises better success. The oil "fog" thus produced by means of compressed air or other medium is very similar to the ordinary oil vapor with the difference, however, that it possesses ordinary room temperature and is much more stable than the latter (a cloud of oil particles so formed will remain suspended in air for several seconds and only gradually disappear.) Condensation is entirely avoided since the oil in effect retains the liquid state. It is of course impossible to prevent some of the oil particles from gathering into drops when striking valve disks and cylinder walls; but intelligent design of the inlet passages can reduce this difficulty to a minimum, a thing which is very difficult to do with oil vapor.

Success should therefore be sought in complete atomizing and spraying rather than in high temperature vaporization. This conclusion is further justified when we consider the subsequent steps in the combustion process. Heated oil vapors soon reach their ignition temperature during compression. This means the use of a lower compression pressure, which, in turn, as has previously been pointed out, leads to lower efficiency and smaller specific capacity of engine. The latter loss is aggravated by the fact that the density of the heated charge is low at the outset. On the other hand, the transformation of the oil "fog" into oil vapor during the compression stroke utilizes some of the heat of compression, hence a higher, that is more efficient degree of compression becomes possible. This advantage is very strongly apparent in the case of alcohol vapor with which, on account of the high specific heat of the water carried and the consequent lowering of the temperature, compression pressures of from 10–12 atmospheres may be carried without pre-ignition. The general axiom "to have the mixture as cool as possible" therefore applies also with equal force to oil engines.

The vaporizer is placed either before or behind the inlet valve. In the latter case it is apt to lose its separate identity. It is heated either externally, by a special heating lamp, or internally by the heat of combustion. As long as the vaporizer temperature remains considerably below the ignition temperature of the mixture, the oil may be vaporized with unlimited air supply, i.e., the mixture may be formed at the same time. Highly heated vaporizers on the other hand should contain little or no air, as the mixture may take fire on the hot walls, consequently the mixture can only be formed beyond the vaporizer proper. It is hardly possible to make a statement of general applicability regarding the superiority of one or the other of these two types of vaporizers. The writer believes, however, that the moderately heated vaporizers will come into use largely for engines using kerosene only, because a colder and more uniform mixture may be obtained from them than from the highly heated vaporizers. For alcohol engines, low vaporizer temperatures always serve the purpose, since alcohol starts to vaporize at about 175° F. and water at 212°. The widespread impression that alcohol requires an especially high vaporizer temperature is therefore without foundation.

Designs of Vaporizers, Atomizers and Heating Lamps:

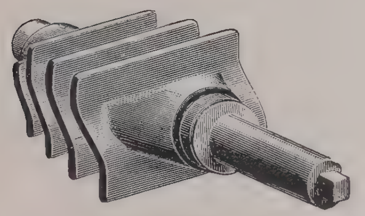


FIG. 382.—Capitaine Vaporizer.

(The liquid fuel enters the tube through a very small opening. The tube itself ends in the combustion chamber, just below the inlet valve.)

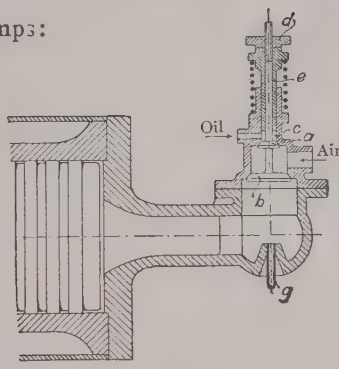


FIG. 383.—Vaporizer, Henriod Schweizer (Lüde Type).

(The inlet valve a has two disks, of which b controls the mixture and c the quantity of oil. By means of screw d and sleeve e the capacity of the space above c may be regulated.)

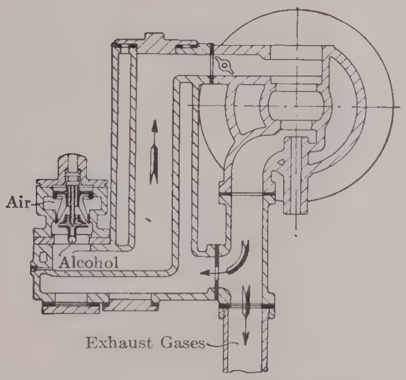
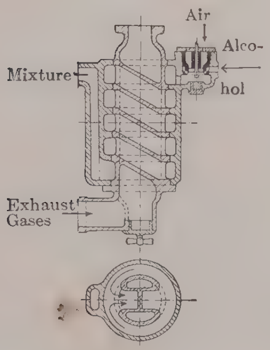
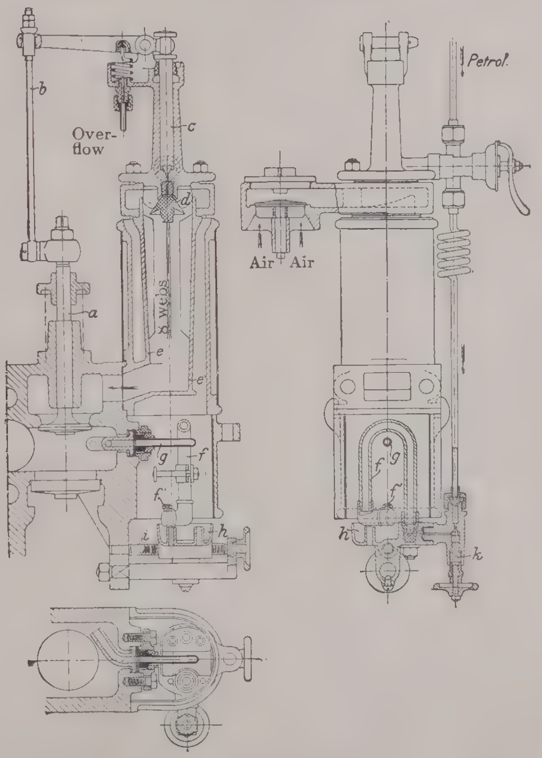


FIG. 384.—Vaporizer, Körting Bros.

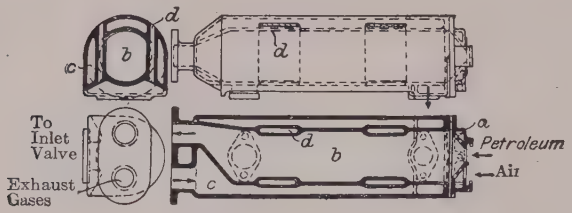


FIGS. 385 and 386.—Dürr Vaporizer.



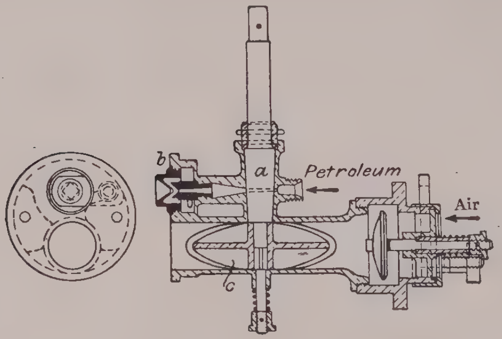
FIGS. 387–389.—Kjelsberg Vaporizer, Ludwig Nobel, Petersburg.

(Inlet valve a operates the fuel valve c through the linkage b. The kerosene admitted by c drops from the atomizer plate d on to the radial webs of the vaporizer e, through whose jacket space e' the hot gases from the hot tube heating lamp f are passed. f' is the burner, h a small basin for pre-heating f', i a screw to adjust the hot zone along the tube g, k the needle supply valve for the burner.)



Figs. 390–393.—Priestman Vaporizer.

(The atomizer (*b* in Fig. 395) sprays into the annular chamber *a*, while the air supply pipe (*c* in Fig. 395) opens into the vaporizer chamber *b*. Jacket space *c* is heated by the exhaust gases, while the passages *d* are heated by a heating lamp at starting.)



Figs. 394 and 395.—Priestman Atomizer.

(The governor, by regulating the position of the plug-cock *a*, controls both the supply of oil through *b* and of air through *c*.)

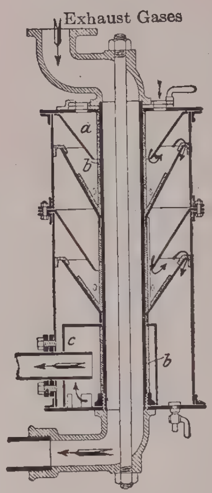


Fig. 396.—Petreano's Carburetor and Pre-heater.

(Air and fuel oil are supplied to the upper space *a*. The oil is absorbed by metallic wicks *b*, which cover the central tube, is vaporized and picked up by the passing air. The mixture passes through chamber *c* to the engine.)

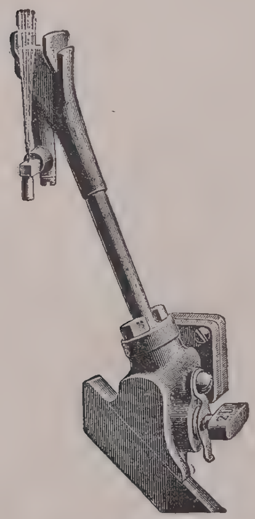
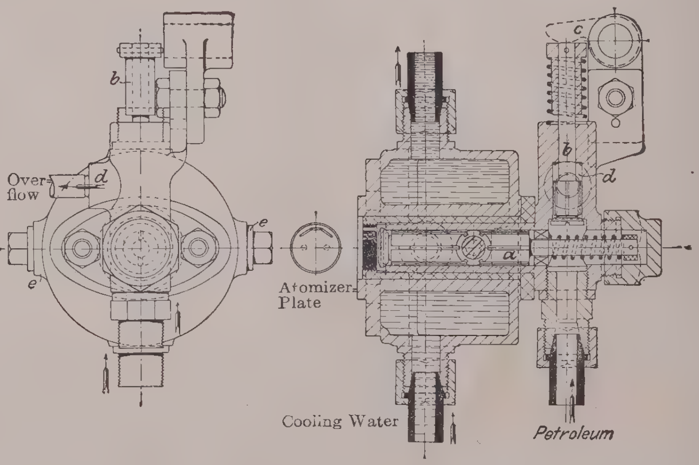


Fig. 397.—Kerosene Heating Lamp, Capitaine.

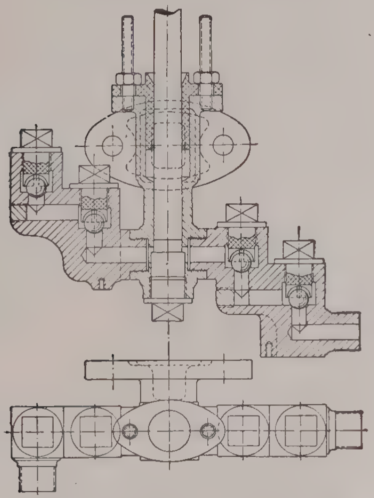


Figs. 398 and 399.—Hornsby Atomizer Plug and Governor Valve.

(The action of the atomizing plug or valve *a* is automatic. The by-pass valve *b* is opened more or less by the governor, through *c*, so that a variable part of the oil furnished by the pump flows back into the reservoir through *d*.)

3. Liquid Fuel Pumps. The minuteness of the quantity of oil to be furnished to each charge requires pumps of very special construction, some types of which are shown in Figs. 400–405.

Designs of Fuel Pumps:



Figs. 400 and 401.—Hornsby Oil Pump.

(The plunger is operated by the inlet valve lever.)

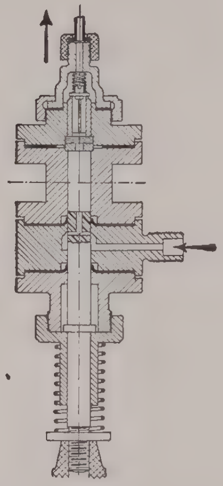


Fig. 402.—Oil Pump, Grob & Co

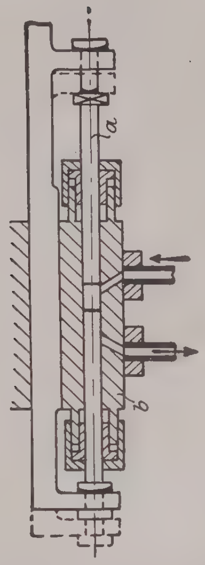


Fig. 403.—Oil Pump, in which the inlet and discharge parts are controlled by the small piston or plunger *a*.

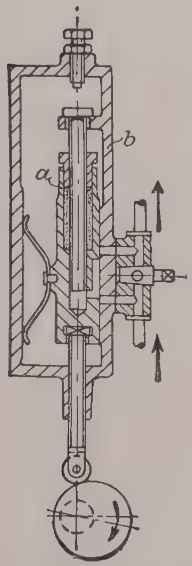


Fig. 404.—Oil Pump, in which the suction and discharge passages are opened and closed by the cylinder *a* sliding up and down in the housing *b*.

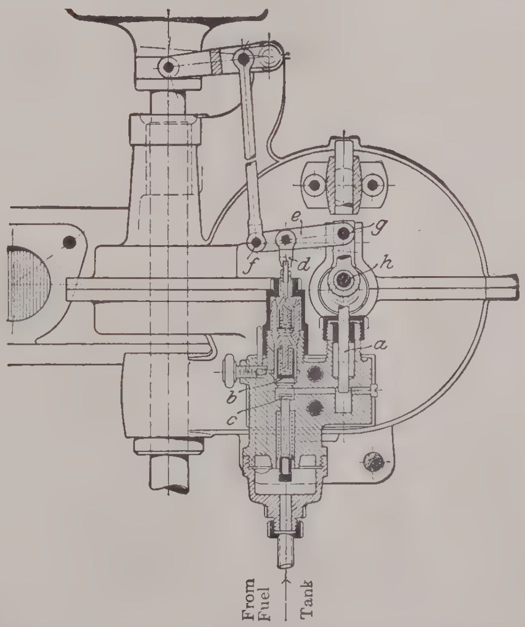


Fig. 405.—Oil Pump and Governor Details, Small Augsburg-Diesel Engine.

(Pump plunger *a* is operated from the lay shaft *h*. The latter, through the guide *g*, also operates the lever *e*, to which the linkage *d* of the suction valve *c* is fastened. Discharge valve *b* is automatic. As the governor varies its position, depending upon the load, the pivot *f* of the lever *e* changes so as to cause the suction valve *c* to close at varying points in the discharge stroke of the plunger *a*, thus controlling the amount of oil delivered to the cylinder.)

In order to avoid too minute dimensions for the piston, the displacement of the pump cylinder is often made from 3 to 6 times that required by the maximum engine capacity. The proper proportion of the mixture is then obtained by allowing the plunger to work idle a part of the stroke, forcing a part of the oil through the suction valve or a special by-pass into the supply reservoir. By changing the instant at which this return is interrupted the quantity of oil supplied the machine can be adjusted to suit the load on the engine and thus to regulate the speed. In small engines the tendency at present is to do away with the use of an oil pump altogether and to draw in the oil by means of the engine piston itself. In such installations the oil reservoir is placed 1.5 to 3 ft. above the point of exit from the supply pipe into the cylinder, unless the oil tank is under air pressure. In order to prevent too great variations in the composition of the mixture when this scheme is used, the hydrostatic pressure on the orifice is often maintained constant by float or overflow valves; that is, it is rendered independent of the level of the supply in the tank. Good fuel pumps, however, insure greater reliability of operation, better atomizing of the oil and the formation of a more uniform mixture. They should therefore always be used in large engines.

E. AUXILIARIES

I. Power Gas Installations

For information regarding fuels and power gases, see Part IV. Concerning the theory of the production of power gas, see Appendix.

The general arrangement of the main parts of a gas-producer installation, that is, of gas producer and purifier or washer, depends primarily upon the properties of the fuels to be gasified. The main points to consider in this regard are the moisture content, proportion of earthy and tar-forming constituents, tendency to coking and clinkering, size of the fuel and the heating value of unit volume. Unless there is some counteracting agency, a high moisture content of the fuel reduces the temperature of the producer in too great a degree, and in this way interferes with the gasification process. Too great a percentage of ash quickly clogs up both generator space and grate, and hence hinders the passage of air and the development of heat. Tar-forming gases cause serious disturbances because the tarry hydrocarbons condense in pipe lines and valves and form soot in the cylinder during combustion. Fuels that show a strong tendency to coke or clinker, and those that are too small sized, interfere with the operation of the producer in much the same way as excessive ash. Large-sized or coarse fuel does not burn with the desired uniformity, offers too little surface for gasification, and allows air, water vapor and carbon dioxide to reach the upper part of the producer without decomposition. Naturally those fuels most free from the disadvantages enumerated above, and especially those low in tar-forming constituents, are commonly used for the production of power gas. The fuels that most nearly fulfil these requirements and are approximately free from tar are hard coal, or anthracite, and coke; we therefore find that most gas producer installations are arranged to utilize these fuels.

The method of producing a suitable power gas from bituminous fuels, especially from the more recent coal formations (lignite and peat) is at present being developed with a great deal of energy, but until now no completely practical and thermally efficient producer has been found. Several installations of this type now in operation have, however, shown a promising beginning. After it had been demonstrated that it was impossible with simple means to continuously and thoroughly remove any considerable quantity of tarry gases from the producer gas leaving the generator, the solution next attempted was to so conduct the gasification of the solid fuel that the tar could be dissociated or broken up into fixed gases inside of the producer itself, and thus to prevent the tar vapor from appearing in the producer gas at all. Tar may be broken up into fixed gases by leading the gas carrying it, or the tar vapor by itself, through a layer of incandescent coke together with air and water vapor. This either burns the tarry hydrocarbons to carbon dioxide (CO_2) and water vapor (H_2O) or breaks them up into carbon monoxide (CO) and hydrogen (H_2). The gasification of the solid fuel and the subsequent fixation of the tar may be carried on either in the same generator or two or more generators may be connected in series above or behind each other, as shown later by several examples.

1. The Power Gas Producers (Generators). Power gas producers are closed vertical retorts in which suitable fuels are burned with insufficient air supply and with the addition of superheated steam. The final product of this operation is a combustible, composite gas containing, besides nitrogen, mainly carbon monoxide (CO) and hydrogen (H_2). In the generator proper, about 85% of the heat of the fuel is transferred to the gas, that is, is recovered. Starting, stand-by periods, incomplete combustion, etc., cause a loss of from 10 to 15% of the fuel. Where separate boilers are used, about 8% of the total fuel is used for the making of steam. Consequently the real efficiency of the generator is only from 60 to 75%, and only in isolated cases higher than the latter figure. One pound of coal gasified usually produces from 70 to 80 cubic feet of producer gas with an average heating value of 135 B.T.U. per cubic foot. Along with the same weight of coal there is introduced about .75 lb. of water, except in the case of suction gas generators, where for the purpose of cooling the grate the quantity used is generally much greater, up to 1.5 lbs. per pound of coal.

Depending upon the method of moving the air-steam mixture through generator and washer, we distinguish three types of producer installations:

(*a*) Pressure gas installations, in which the air and steam are forced through the fuel column by means of special blowers, and the steam is usually generated in a separate boiler.

(*b*) Suction gas plants in which the suction of the engine piston draws the steam and air mixture through the charge in the generator, and in which the required steam is usually produced by recovering some of the heat of the generator.

(*c*) Combination plants so constructed that a fan or blower draws the gas out of the producer and forces it towards the engine.

Suction gas plants have almost entirely displaced the older pressure installations for gas engine work. They do away with the use of a separate boiler with its attendant disadvantages (increased fuel consumption, greater cost of attendance, higher capital cost, increased floor space required, insurance, inspection, etc.). Further it is possible to clean the grates in suction gas plants without interrupting regular operation, and the entire installation is more simple as well as cheaper both in construction and operation. Since the rapidity of gasification depends directly upon the suction of the engine piston, the quantity of gas made will vary directly with the amount required by the engine. This automatic regulation makes the use of gas holders or pressure regulators superfluous, and thus eliminates one of the least desirable features of the pressure gas plant.

Operation with suction gas does not in itself possess any serious disadvantages, where real difficulties crop up they are usually due to poor construction or wrong management. The partial vacuum existing in the system may at first sight appear a serious drawback, but examination shows that with proper construction the pressure can be kept to within from 4″ to 8″ of water of the atmospheric pressure, which is less than the regular variations in the barometric readings. Only with very poor construction or bad operating conditions does the vacuum reach a degree (40″ or more of water) that markedly affects the specific capacity of the engine. The misplaced apprehensions concerning the danger of the vacuum existing in suction systems are directly responsible for the fact that the scrubbing and washing apparatus for such plants are made much smaller and less complete than experience has shown is necessary for pressure installations. As a matter of fact, suction gas may be just as thoroughly cleaned as pressure gas, but, in place of simply transferring and utilizing old designs of working apparatus it is necessary to employ washers adapted to the particular purpose. With this precaution it should be possible to satisfactorily clean suction gas without materially affecting engine capacity.

The limitations in the use of suction gas for engines sometimes found where the gas is also used for other purposes (heating furnaces, etc.) have been eliminated by the use of combination systems or by drawing the gas intended for heating purposes by means of blowers and forcing it to the furnaces.

There remain to be considered two things which nave been used as charges against the suction gas principle, serious enough to arouse insurance companies and to cause government supervision in some quarters—that is, dangers due to explosions and to poisoning. Cool and deliberate consideration of all the facts in the case will, however, show that suction gas plants not only warrant greater safety against either one of these accidents than pressure gas producers left so long without molestation, but further, that a number of serious and inexcusable irregularities in attendance and maintenance will have to concur in order to produce conditions that will lead to really dangerous poisoning or explosion accidents in suction plants.[1]

At the same time it cannot be denied that there are certain further problems in the development of suction gas plants. The number of different kinds of fuel that may be properly gasified in them is at present very small, and even for these there are features that call for improvement. The first of these is the present method of producing and introducing the steam because the ordinary vaporizers all labor under the disadvantage of having too great a water space. The large amount of water in the vaporizer (superheater) seriously increases the time of starting, since the water must be brought to the boiling point before the gas has the normal composition. As long as the water does not steam, the gas will lack its main heat-carrying element—hydrogen. The engine is started with difficulty and only after a considerable length of time may it be fully loaded. After the engine is stopped the steam finds its way from the vaporizer into the room, and causes trouble through condensation, rusting, etc. Specially bad, however, is the fact that with vaporizers having a large water space the composition of the gas cannot be controlled with any degree of certainty.

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[1] To say a few words in explanation of this statement it may be noted that the gas escaping through small leaks in pressure gas plants, even when in considerable quantities, is not noticed by the attendants, because carbon monoxide gas is odorless and colorless. The danger of poisoning and explosion in this case may therefore appear very suddenly and without warning. In suction-gas plants this condition cannot occur. Small leaks may of course cause a decrease in the quantity of gas in the mains, which is in most cases quickly noted by a decrease in the engine power, but to produce explosive mixtures requires such serious defects in piping that the condition is hardly conceivable in practice, because the ratio of air to gas must be in the neighborhood of 1:1. On the basis of such facts the suction-gas producer had better be left unfettered with any government regulations in its development—it is no more dangerous than the ordinary heating stove.

A change of the water supply to the vaporizer has no effect, and very often produces the very opposite from the result intended. If, for example, the hydrogen content of a lean gas is to be raised, the attendant naturally increases the supply of water to the vaporizer. This, however, is immediately followed by a cooling of the water in the vaporizer as well as the vaporizer walls. The result is a reduction in the quantity of steam evolved and the gas made is even poorer in hydrogen than before. If, on the other hand, the engine shows by pounding and other signs that the hydrogen content in the gas is too high, the only remedy available is to cut down the water supply to the generator. The inevitable result of this, however, is that the evolution of steam becomes stronger than it was before and will remain so until the vaporizer goes dry.

These difficulties are avoided if there is no water storage capacity whatever in the vaporizer, and only as much water as can be used per suction stroke of the engine is furnished to the generator. In case the engine is governed by throttling, the periodic introduction of water should be so regulated that the resulting gas shall always have its best composition for all loads on the engine. This requirement, important alike from the standpoint of economy and reliability, is to-day fulfilled by only a few of our suction gas plants, while all other installations operate normally only in the neighborhood of certain loads and are likely to give trouble when this load is materially changed.

The usual construction of tube vaporizers should also be quickly changed to something better. In them, short tubes are usually held between inelastic tube sheets and constantly exposed to the effects of widely varying temperatures. The result is that they are often the cause of costly repairs and troublesome interruptions in operation. On account of the sudden decrease in the gas velocity and the change in direction of flow, dust is deposited thickly in the end spaces of the vaporizer above and below the tube sheets, not only decreasing the efficiency of heat transfer, but after a time seriously hampering the flow of gas. Besides this, the tubes increase the distance the dirty gas has to travel and their surfaces are apt to become coated with deposits of dust and tar, more serious in this place because of the reduction of heat transfer and the difficulty of proper cleaning. In the smaller installations the generator vaporizer may be placed directly in the top of the producer; in larger plants its proper place is outside, but so close to the shell of the producer that the distance from the producer to the scrubber shall be as short as possible, and so constructed that there shall be no chance for deposits of tar or dust.

The vaporizer should be heated only by means of the waste heat of the generator or the sensible heat of the gas. To use the exhaust gases of the engine for this purpose is from the thermal standpoint mostly folly,[1] and in many cases even absolutely wrong because it increases the cost of the plant as well as the back pressure on the engine. The advice to use a part of the jacket water to feed the vaporizer is equally ill-considered because for every pound of coal used the engine requires from 60 to 90 lbs. of jacket cooling water, while only from .7 to 1.5 lbs. of water, i.e., only about 1/70 of the former amount is fed to the generator for the same amount of coal.

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[1] Proof. Assuming a very good efficiency for the generator, say 75%, we find for every pound of coal (14 300 B.T.U.) gasified in round numbers 3500 B.T.U. are discharged in the shape of radiant heat and sensible heat in the gas. This heat is quite sufficient to evaporate about 3 lbs. of water, that is, more than twice the maximum amount that any normal generator would require. Any sensible heat that is not removed from the gas before it enters the washing apparatus is in most cases absolutely lost. To recover the sensible heat of the gas by pre-heating the air for the generator is possible only in limited degree, even in large plants fitted with special pre-heaters. The ordinary constructions of this type at any rate show no added advantages both on account of the increased suction and exhaust friction losses and because of the rapid destruction of grates, etc.

Designs of Gas Producers. (Figs. 406–431, producers for anthracite and coke, Figs. 432–440, producers for fuels carrying tar.) For data concerning practical operation, see Part III.

(*a*) *Pressure Gas Installations*

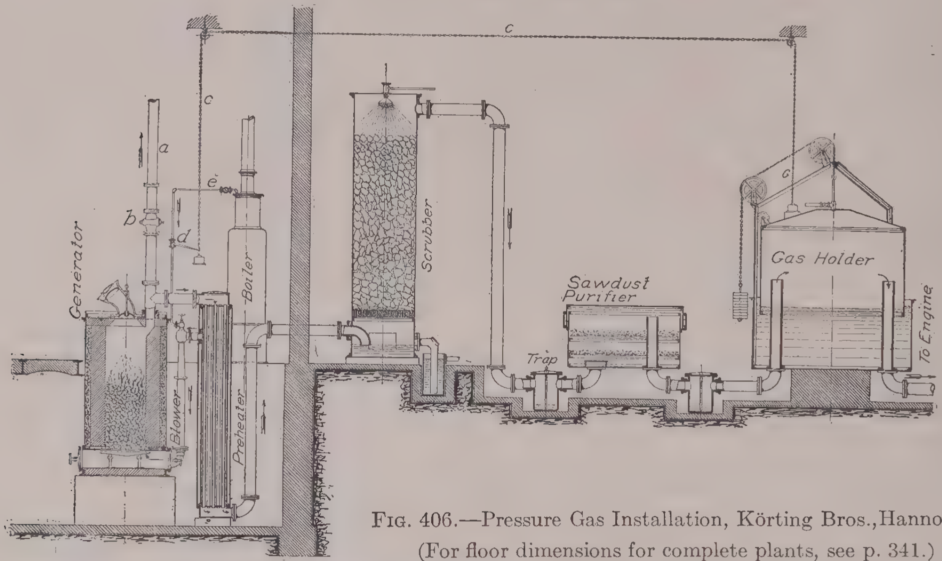


Fig. 406.—Pressure Gas Installation, Körting Bros., Hannover.

(For floor dimensions for complete plants, see p. 341.)

The poor gas made during starting escapes through valve *b* and purge pipe *a*. During this time the producer operates like a common stove. As in the case of a stove, the introduction and firing of the first fuel is done through the open ash pit doors. After sufficient fuel is added, these doors are closed and the steam blower is started. In case the natural draft available is not sufficient to properly start the fire, a small steam ejector in pipe *a* may be used to increase the draft. As soon as the flame at the try-cock shows dark red, not blue, the gas is of proper quality, the valve *b* is closed and the plant is ready for operation.—The cleaned gas on its way to the engine passes through a regulator (marked "gas holder"). The rising and falling of the bell of this regulator opens and closes the steam supply valve *d* by means of the chain connection *c–c*, thus controlling the amounts of air and steam furnished the producer.—According to the builders, the average composition, in per cent by volume, of the gas made in producers of this type is as follows:

Hydrogen, 18%; carbon monoxide, 26%; hydrocarbons, 2%; Carbon dioxide, 7%; and nitrogen, 47%. The efficiency of the producer is from 80 to 82%.

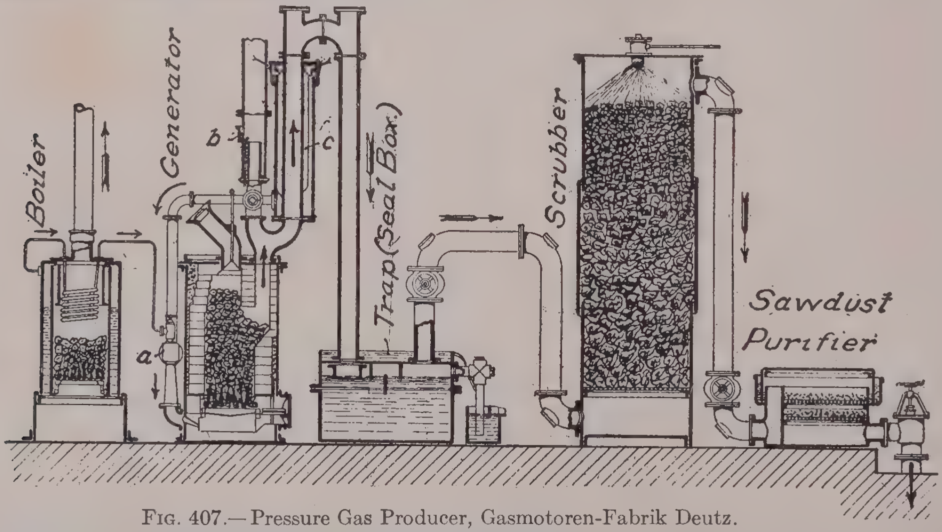


Fig. 407.—Pressure Gas Producer, Gasmotoren-Fabrik Deutz.

The steam for blower *a* is superheated in the coil located in the upper part of the boiler. During starting the gas may be tried at opening *b*. The air injected by blower *a* is pre-heated by passing through the economizer *c*.

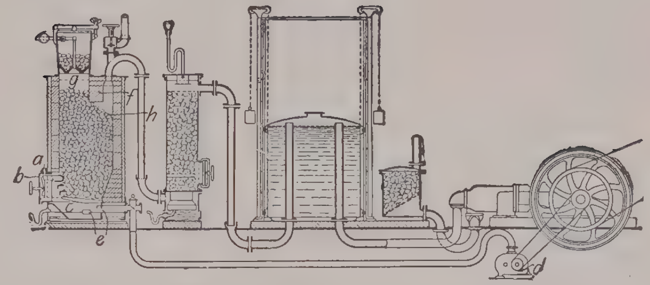


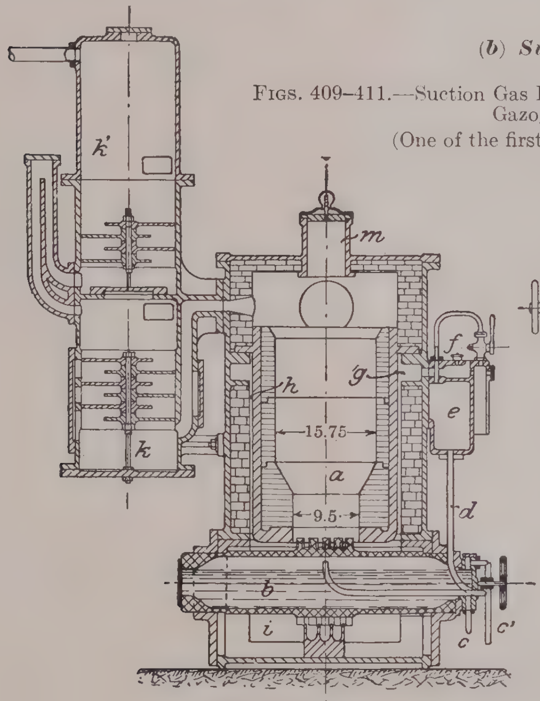
Fig. 408.—Pressure Producer Gas Installation without Boiler, Lencauchez.

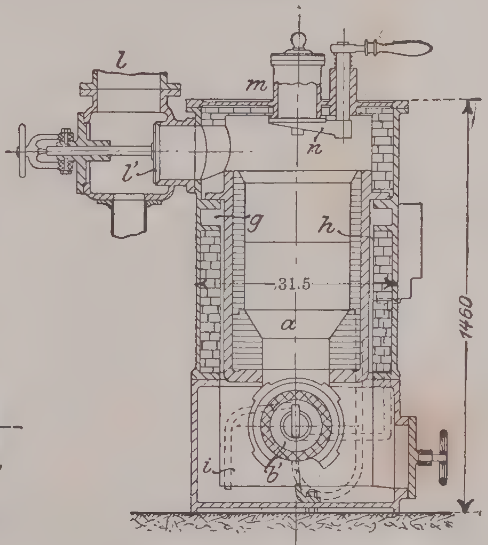
Water is sprayed through pipe *a* continuously against the ribbed plate *b*. It drips from here into the cast iron ash-pan *c*, vaporizing in its course. At the same time the positive blower *d* blows air, slightly compressed, through several radial channels *e* into the ash pit *c* where the air picks up the vapor and the mixture then passes through the producer. A fire brick arch *f* divides the charging space *g* from the gas outlet space *h*. The former is filled with fresh fuel which slowly moves downward as demanded. Since the blower is driven by the engine and no gas can be made unless it operates, the gas holder should always contain sufficient gas when the engine is stopped.

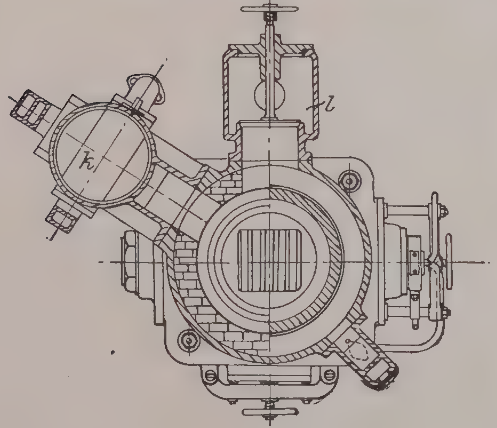
(*b*) *Suction Gas Plants*

Figs. 409–411.—Suction Gas Plant, 25 H.P., made by Société des Moteurs Gazogène Bénier, Paris.

(One of the first type of suction gas producer.)







The lower end *a* of the producer is closed by a hollow cylindrical grate *b* which in the interior acts as the vaporizer chamber. The latter is constantly supplied with water through *c*, any excess flowing out through *c'*. The steam formed reaches the chamber *e* through *d*, and finally finds its way into the mixing chamber *f*. On account of the sucking action of the charging pump of the engine, air also enters *f* from the outside, and the air-steam mixture formed is drawn into the space *g*, down through the annular space *h* and thus reaches *i*. Here it surrounds *b* and then enters the producer proper. The gas made passes out through two washers *k* and *k'*.

The grate may be rotated on its axis, the construction is plain from the figure. Purge pipe *l* is closed by a disk valve *l'*. Charging hopper *m* is closed at the bottom by a slide *n*, which may be shifted to one side.

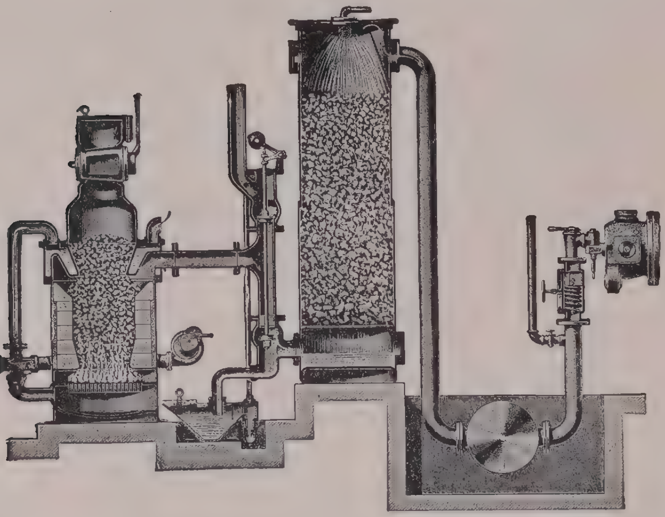


Fig. 412.—Suction Gas Plant, made by the Gasmotoren-Fabrik, Deutz.

The vaporizer, in which the water is kept at the same level by means of an overflow, surrounds the top of the producer. On the suction stroke of the engine air enters the vaporizer through the elbow at the right, and is saturated with water vapor. The air-steam mixture then passes through the pipe at the left, under the grate and through the incandescent column of coal, in which the well-known reactions resulting in producer gas take place. The gas next passes through the valve **5**, up through the wet scrubber, through the gas reservoir and on its way to the engine finally through a dry scrubber (wire brush).

The double lock construction of the charging hopper is plain from the illustration. On starting the producer, valves **1** and **2** in the pipe at the left of the producer are closed, while valve **3** is opened. Next to the scrubber, valve **5** is closed, at the same time opening valve **4** leading to the purge pipe. The hand blower then forces air through the producer and up the purge pipe. During stand-by periods, valves **1**, **3**, and **5** are closed, while, on account of the draft through the purge pipe, a little air passes in at **2**, keeping the fire alive. The check valve in the pipe below valve **1** is intended to prevent any striking back of hot gas when the change from normal operation to stand-by is being made. Any "overpressure" of gas that may form at this particular moment is taken care of by an equalizer pipe leading from the seal-box to the purge pipe. (The construction of the larger Deutz plants does not quite agree with Fig. 412.)

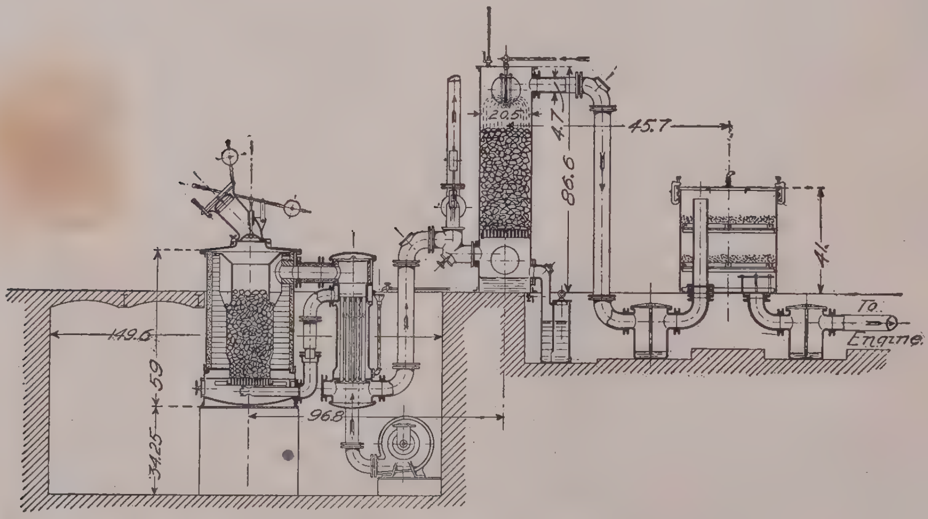
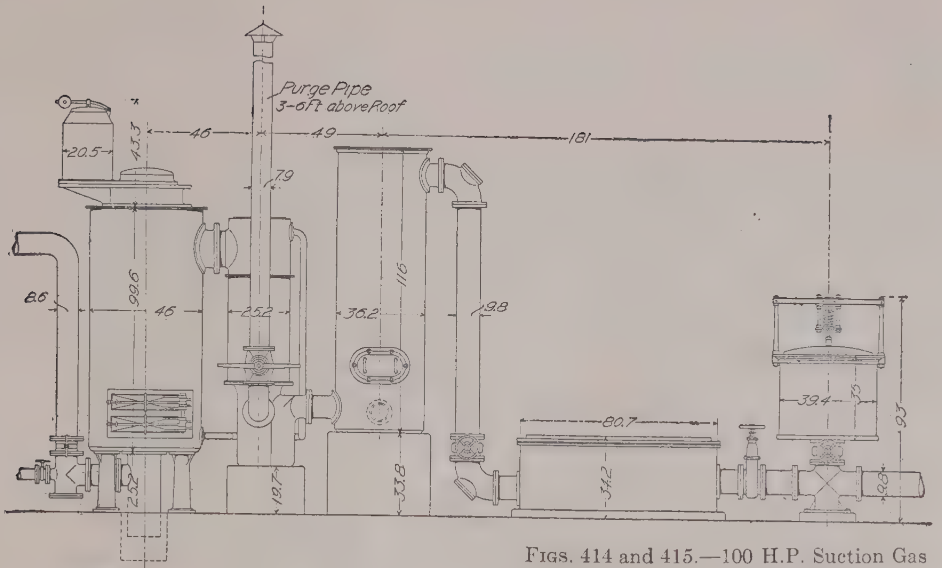


Fig. 413.—30–35 H.P. Suction Gas Installation, Körting Bros.

The tubular vaporizer, located at the side of the producer, is heated by the outgoing gases. The purge pipe branches off beyond the vaporizer, so that the latter is also being heated during the starting period. The starting fan blows into the gas space of the vaporizer. (Type of producer for bituminous fuels, see Fig. 433.)



FIGS. 414 and 415.—100 H.P. Suction Gas Plant, Jul. Pintsch, Berlin.

Cover and filling hopper are supported on a sector-shaped plate which may be swung about a pivot at the side. FIG. 414 shows the operating position. The producer is charged by swinging the hopper over the center opening, see Fig. 419. The small branch pipe near the bottom (at the left) of the air supply pipe serves to furnish air under slight pressure for starting.

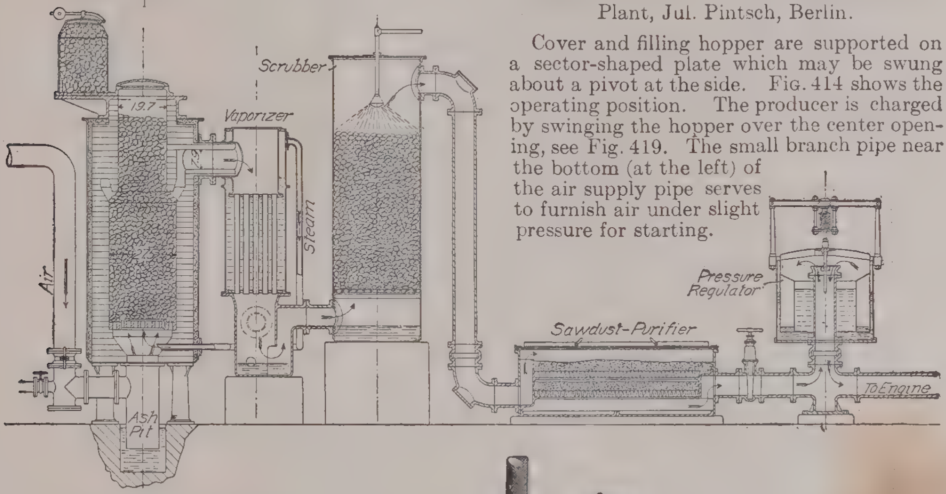
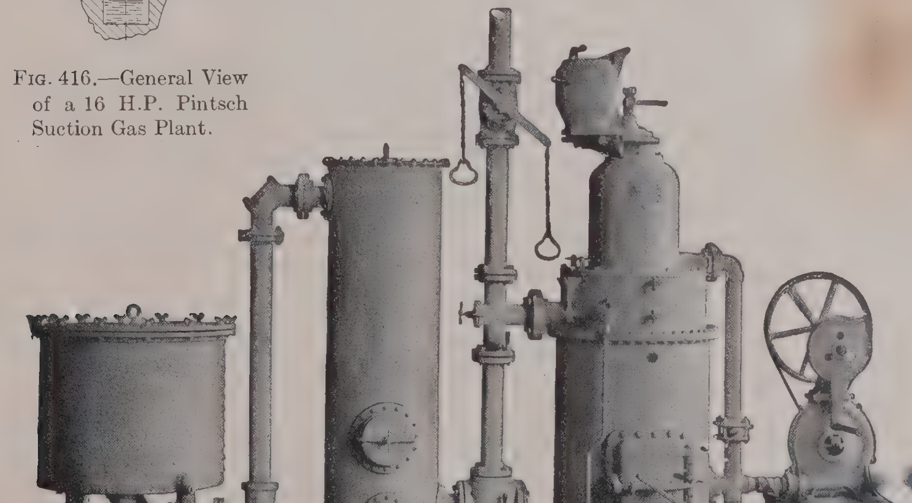


FIG. 416.—General View of a 16 H.P. Pintsch Suction Gas Plant.



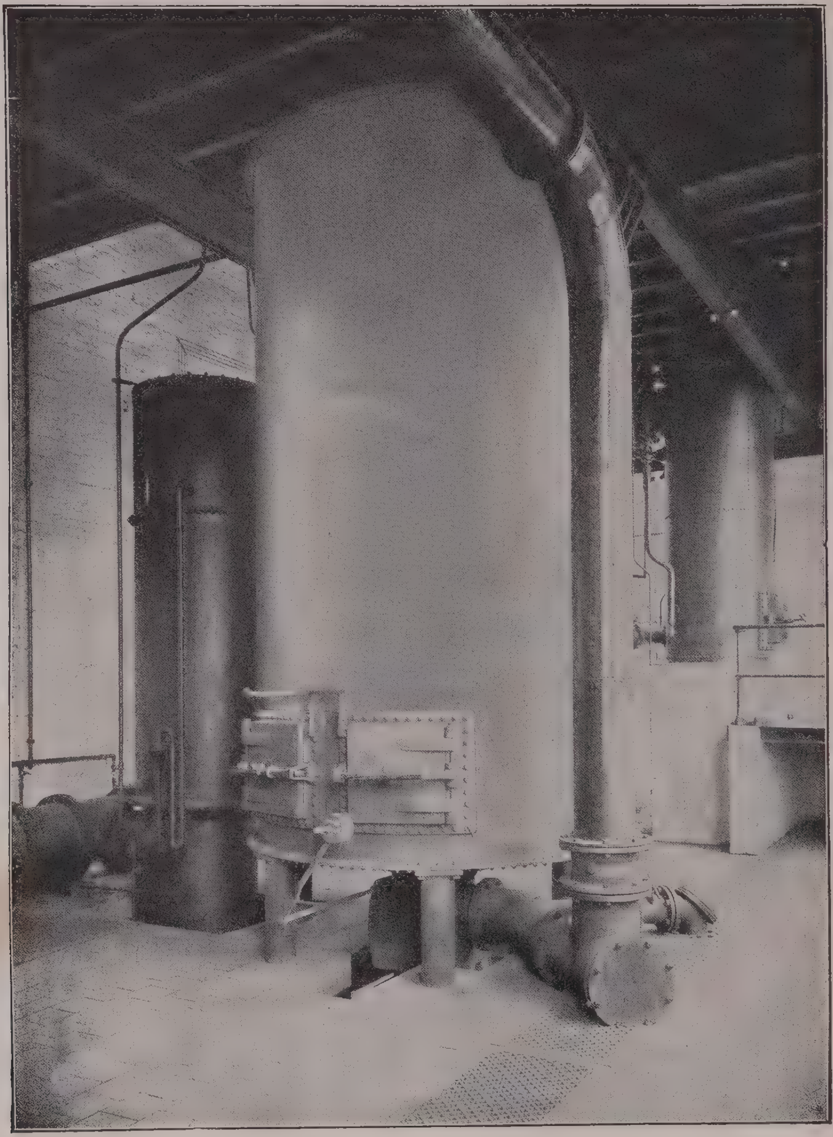


Fig. 417.—Suction Gas Installation, Pintsch.

(2 Producers, 600–800 H.P.)

The gas is led through one 20″ main pipe to three twin cylinder engines, each rated at 200 B.H.P. A Pintsch pressure regulator is connected in each branch main to the engines. The apparatus in the background is a small water-gas producer (1500 cu.ft. per hour), the gas being used for heating purposes in the works.

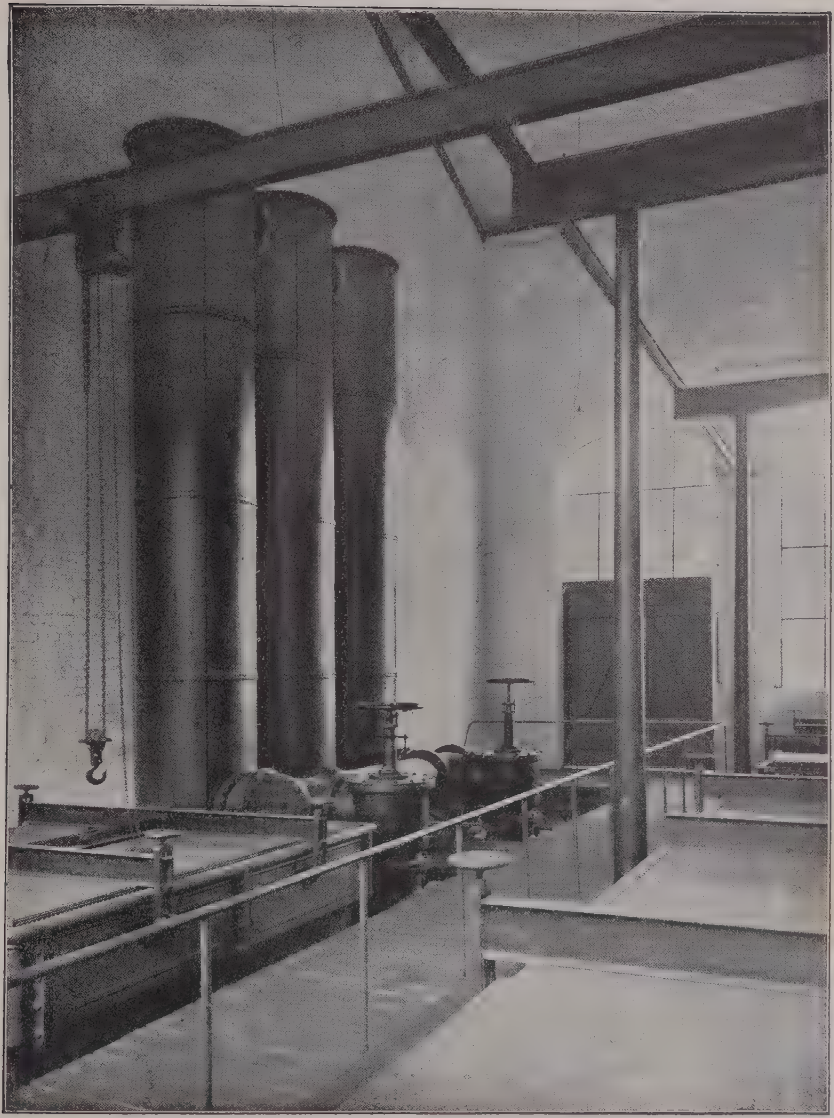


FIG. 418.—Scrubbing Apparatus, 600 H.P. Pintsch Suction Gas Installation.

The picture shows in the left foreground the dry purifier for the water gas. Directly behind this is located the cylindrical scrubber for suction gas between two cylindrical coolers. The latter were here employed because the otherwise necessary cooling water was hard to obtain. At the right are two dry purifiers, 17½ by 21 ft. floor-space, for suction gas.

At the right, in the background, is located a small tubular boiler for the water-gas plant. The charging mechanism is seen directly below the telescope purge pipe (a construction now abandoned). The position of the charging hopper on the producer top may be changed by means of the crank mechanism shown. During ordinary operation, the hopper is shifted to one side (as shown in Figs. 414 and 415) and is only brought over the producer when charging.

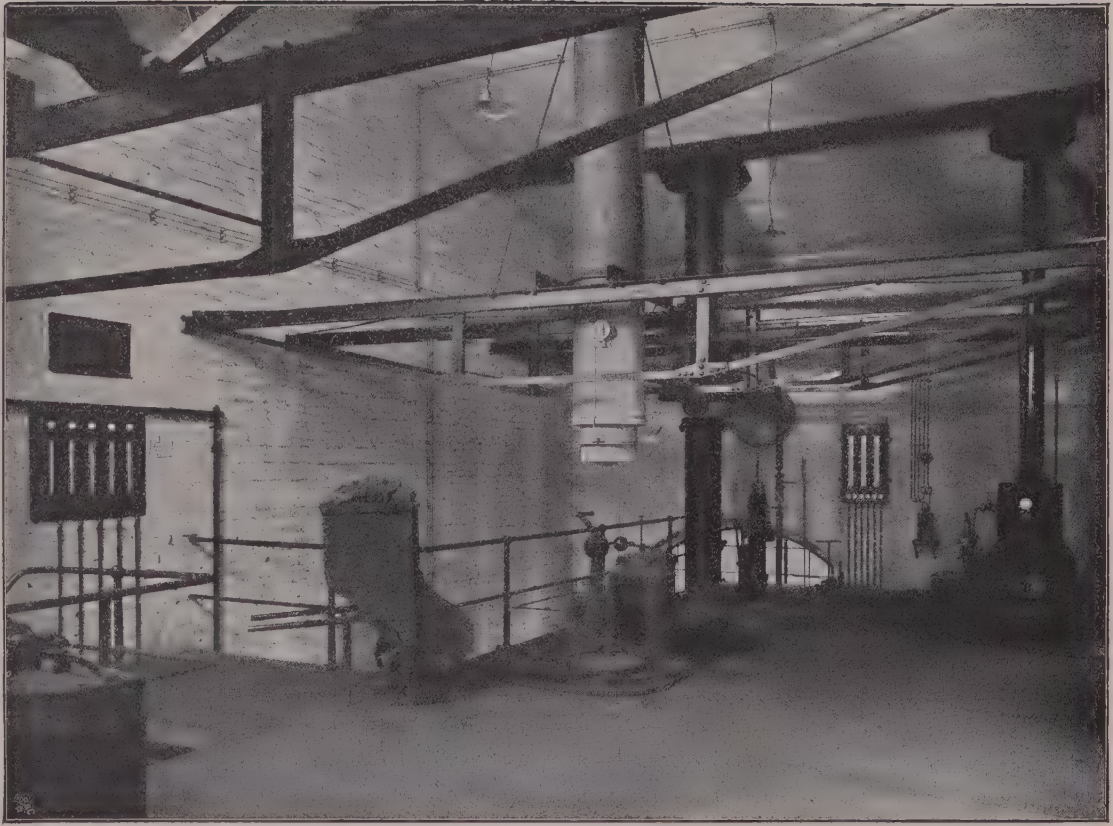
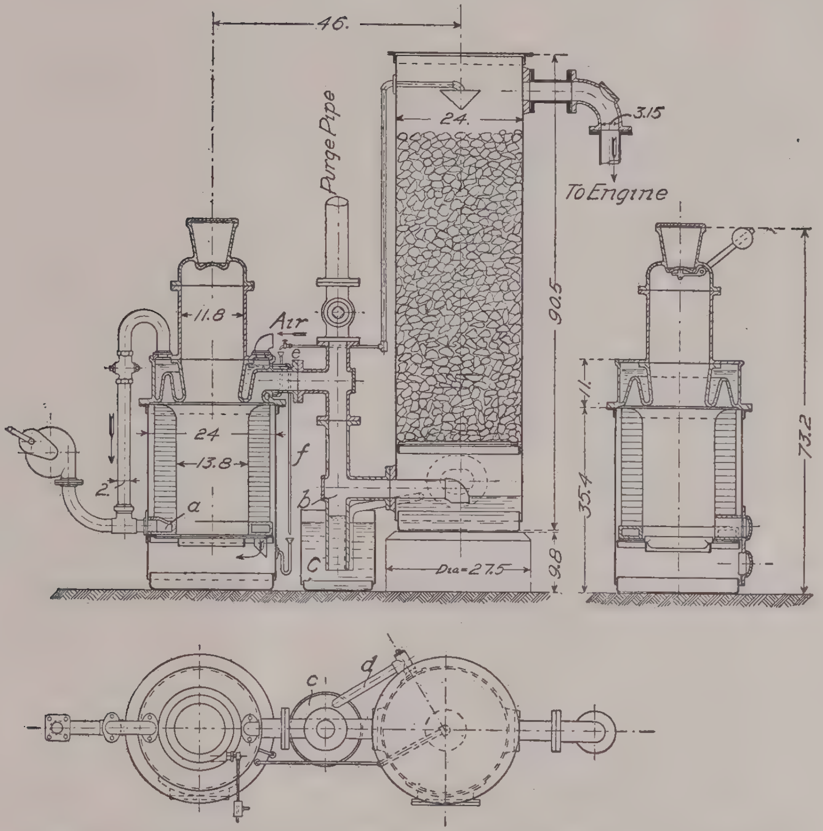


Fig. 419.—Charging Floor of the 600 H.P. Pintsch Suction Gas Installation, shown on pp. 276 and 277.

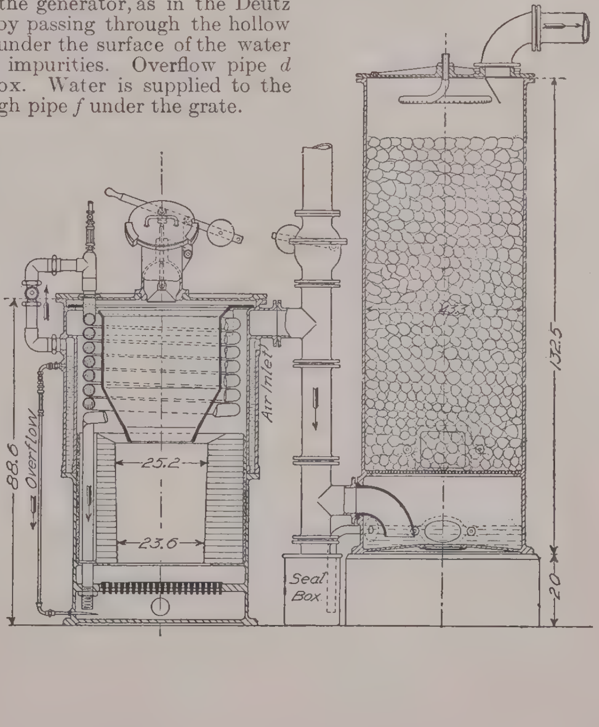


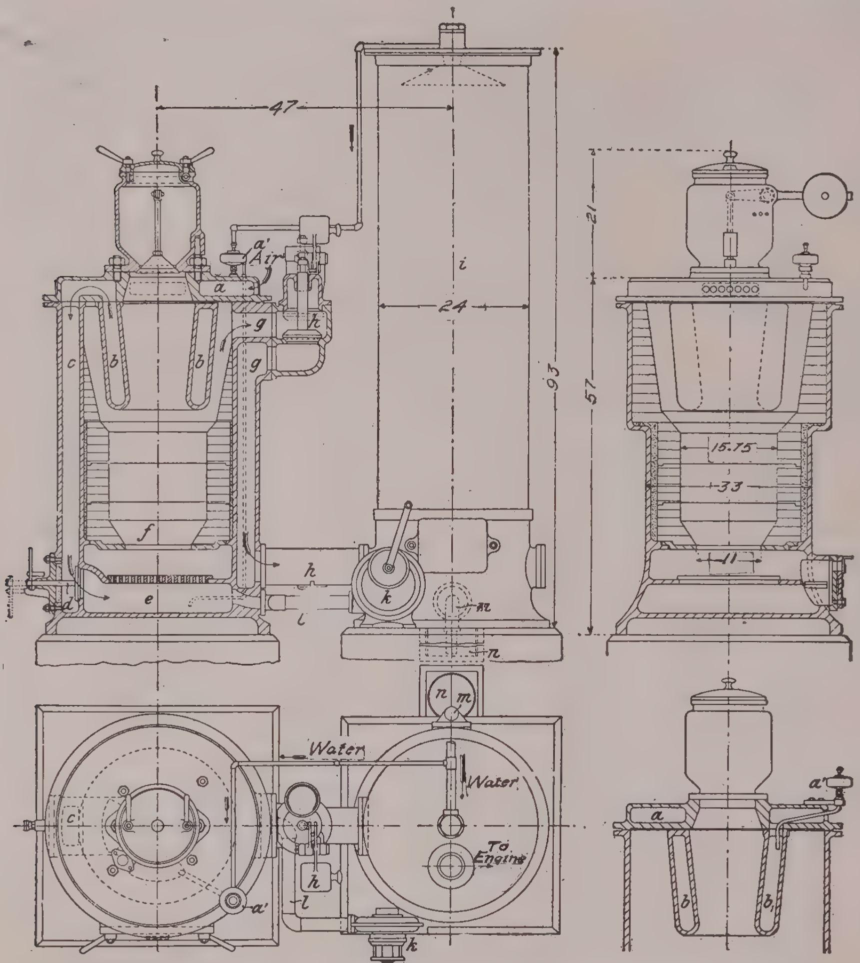
FIGS. 420–422.—25 H.P. Suction Gas Installation, Wiedenfeld & Co., Duisburg.

The vaporizer is located around the top of the generator, as in the Deutz design. The air-steam mixture is superheated by passing through the hollow circular grate support *a*. The gas main *b* dips under the surface of the water in the seal-box *c*, to throw down the coarser impurities. Overflow pipe *d* from the scrubber discharges into this seal-box. Water is supplied to the vaporizer through *e*, any overflow is led through pipe *f* under the grate.

FIG. 423.—100 H.P. Suction Gas Plant, Dunker & Spielter, Hannover.

The jacket space of the producer is used as the vaporizer, a special superheater coil surrounding the top of the producer. At starting a small amount of water is introduced directly into the coil, to obtain sufficient steam from the outset. Only a part of the air used passes through the coil, the rest entering the ash pit directly without being mixed with steam.





Figs. 424–427.—30 H.P. Güldner Suction Gas Installation.

(Cuts show the plant as arranged for starting up.) For floor space required for complete plant, see p. 356.

The producer has no vaporizer containing any quantity of water; air is pre-heated in the top space *a*, then saturated with steam through the action of the supply mechanism *a′* (see Fig. 428). The mixture then flows through superheater *b*, passage *c* and safety valve *d* into the ashpit *e*. The lining *f* is cone shaped at the bottom. The gas passes through double valve *h*, passages *g* and *h* into the wet scrubber *i*; starting fan *k* blows through pipe *l* into the ashpit. Overflow *m* of the wet scrubber discharges into the sealbox *n*.

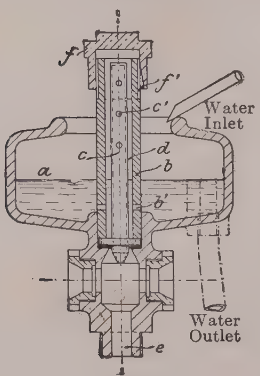


Fig. 428.

Action of the Water Supply and Regulating Valve, Fig. 428: At the center of the water pan *a* there are two concentric tubes *b* and *c*, forming an annular space *d* between them, which is in communication with *a* through openings *b′*. As long as the apparatus is at rest, the water level in *d* will be at the same height as in *a* and stand below the overflow openings *c′*. But on the suction stroke of the engine, the vacuum produced in the vaporizer chamber *a* of the producer, Fig. 424, communicates itself through *e* to the interior of the tube *c*. This causes the water column in *d* to rise, some of the water overflowing through *c′* and so into the vaporizer. The quantity of water delivered increases with the suction, and is hence adjusted to the load on the engine, keeping the hydrogen content of the gas always at the most favorable point. The quantity of water delivered may also be regulated by hand by turning the cap *f* so that some air will be admitted through groove *f′*, thus partially neutralizing the suction.

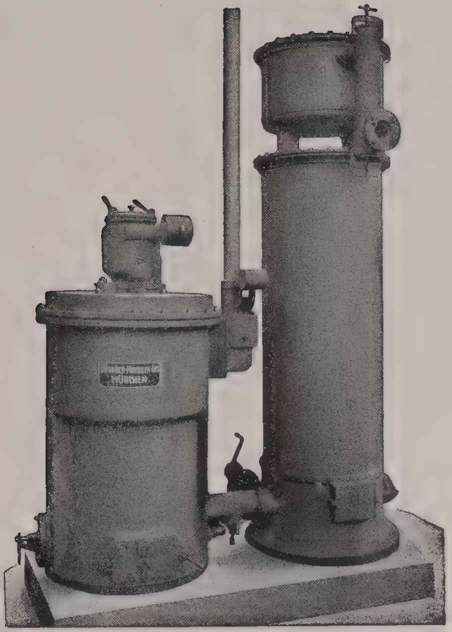


FIG. 429.—General View of a 50 H.P. Güldner Suction Gas Plant.

(Producer, scrubber, dry purifier, and tar extractor.)

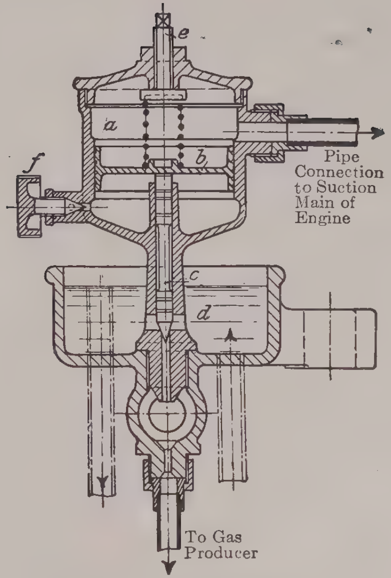


FIG. 430.—Water Supply Valve, Schweiz. Locomotiv- und Maschinenfabrik, Winterthur.

Piston *b*, fitting lightly in cylinder *a*, is raised every time the suction of the engine creates a partial vacuum in the gas main and in the space *a*. This action raises the needle valve *c*, which then momentarily opens the outlet *d* to the producer. The regulating screw *e* serves to adjust the tension of the spring and thus controls the lift of *b*.

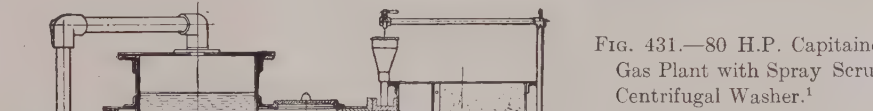


FIG. 431.—80 H.P. Capitaine Suction Gas Plant with Spray Scrubber and Centrifugal Washer.[1]

Tube vaporizer, shaking grate, grate bars of trough pattern filled with ash for protection against heat. Preliminary cooling and washing of the gas in a scrubber with water supply at the bottom. Main scrubbing in a centrifugal washer operated by the engine at a speed of about 100 r.p.m. Centrifugal action is supposed to take out tar and traces of sulphur as well as dust and water.

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[1] Lecture given before the VI Regular Convention of the Schiffbautech. Gesellschaft.

(c) *Gas Producer Installations for Fuels Carrying Tar*

About the only grade of tar-carrying (bituminous) fuels that is used to any extent to-day for the manufacture of power gas is the briquetted soft coal[1] or lignite. These briquettes, on account of their uniform size and their comparatively low content of water and ash, are fully as suitable for this purpose as anthracite or coke, provided of course that the tar is taken care of. If the tar is removed by means of washing and scrubbing apparatus outside of the generator, it is even possible to use the ordinary type of producer. This subsequent separation of tar from the gas, however, soon causes operative difficulties with the ordinary type of scrubbers, and, outside of the fact that the rotary washers require power and that the odor of the scrubber water is in many instances very obnoxious, the method possesses the disadvantage, from the economic standpoint, of wasting the heat contained in the tar-forming hydrocarbons. The existing special constructions of soft coal (brown coal) gas producer installation therefore, in general, all endeavor to fix the tarry gases in the producer itself. The general method of doing this is to collect the gases of distillation and to lead them up through the incandescent bed of fuel above the grate, either simply breaking up the tar or burning it by supplying air, in which case the heat of combustion enters the general gasification process. This method of operation has been comparatively successful in properly designed installations; it is, however, still beset with many practical difficulties. The starting of such generators is a rather complicated process, for which reason, rather than let them burn down, they are banked even for periods covering several days. During operation such producers under certain conditions exhibit a troublesome variation in the position (traveling) of the incandescent zone, in which case the tarry gases are apt to escape undecomposed.

In the design of brown coal or lignite generators, the generally very high percentage of water in this kind of fuel should not be left out of account. This may be so high that the generator requires no further water supply during operation. This does not mean, of course, that the vaporizer may be entirely dispensed with in the common types, but the fact calls for wide range of possible adjustment in the quantity of steam made or that introduced to the generator.

Soft coal gas producers are only in a few cases purely suction gas generators. In general the friction losses and resistances in generator and washing apparatus are too great to operate the installation on this principle, and fans are used to draw the gas through the producer and to force it to the engine. It is quite usual in this construction to make the fan act also as a part of the washing apparatus by the introduction of a spray of water which is afterward drawn off from the lowest point of the fan housing together with the dust picked up.[2] Bituminous coal power gas installations, less simple in their construction than anthracite or coke plants, also cost somewhat more in regard to maintenance and attendance than the latter and their use consequently offers but little advantage for the smaller installations.

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[1] TRANSLATOR'S NOTE. In this country, briquetted coal is as yet not used to any extent; where soft coal is used it is usually in the raw state.

[2] Fan washers have been frequently used for a number of years, as, for instance, in the operation of gas engines with blast-furnace gas, consequently their use can no longer be restricted by any patent rights. Recently, however, Theissen has claimed that the use of blowers working in this manner with water injection constitutes an infringement of his patent rights, and makes their further use subject to his license. This controversy has an important bearing upon the gas-engine industry in general, and to help clear matters the writer has just caused the Gasmotorentechnik (April, 1905), to start a public discussion of the question.

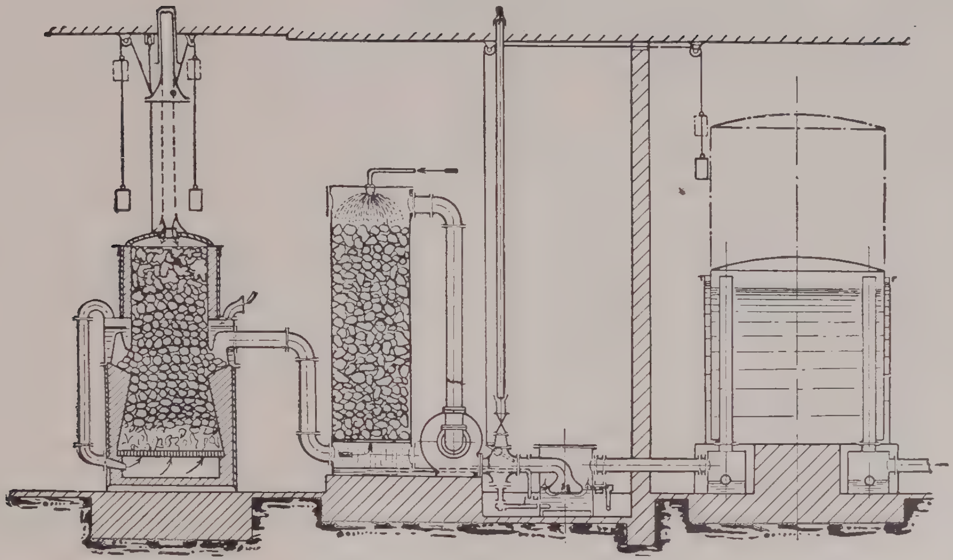


FIG. 432.—Double Zone Generator for Brown Coal, Gasmotoren-Fabrik Deutz.

This type of generator, in addition to the combustion zone just above the grate, has another at about the level of the vaporizer, near which is also located the gas exit. The fresh fuel fired on top of the upper zone is first subjected to distillation. The gases formed are drawn downward through the incandescent layer of coked fuel by the action of the exhauster connected in beyond the scrubber. This serves to convert these hydrocarbon gases into permanent gases. Air is drawn at the same time through openings in the cover, thus causing a partial gasification of the coked fuel near the top, which action furnishes the heat necessary for distillation and coking. The coked fuel slowly sinks downward, not all being gasified in the upper zone, and thus maintains the combustion zone above the grate. The action of the exhauster causes air to enter the vaporizer and the air-steam mixture so formed flows in under the grate and through the lower zone resulting in the production of ordinary producer gas. The cleaning apparatus consists of a wet scrubber, a centrifugal exhauster with water injection and a settling chamber. The bell of the gas holder controls a by-pass valve on the exhauster and thus regulates the draft on the producer.

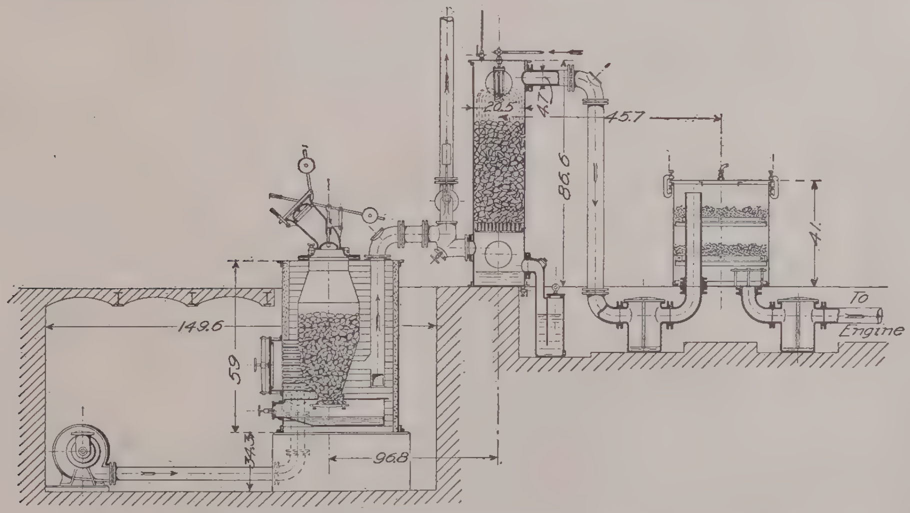


FIG. 433.—Brown Coal Gas Producer Installation, 30 H.P., Körting Bros.

Besides the ordinary flat grate there is furnished a second inclined grate at the side, to aid in starting and cleaning the fire. The operation is otherwise much the same as for Fig. 432, the gas being taken off about half way up instead of at the top. The tarry hydrocarbon vapors formed at the top are sucked back through the already coked layers of fuel and, meeting the current of air and steam coming up from below, are either burned or fixed. A special vaporizer is not required in this case on account of the high moisture content of the fuel. The blower at the left is used at starting.

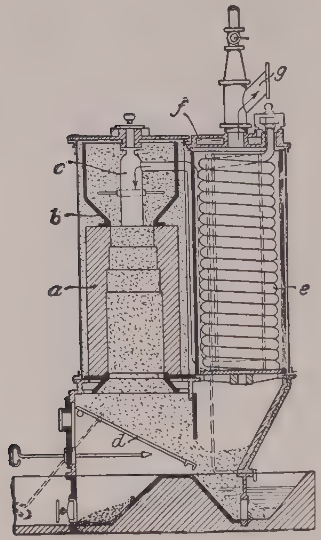
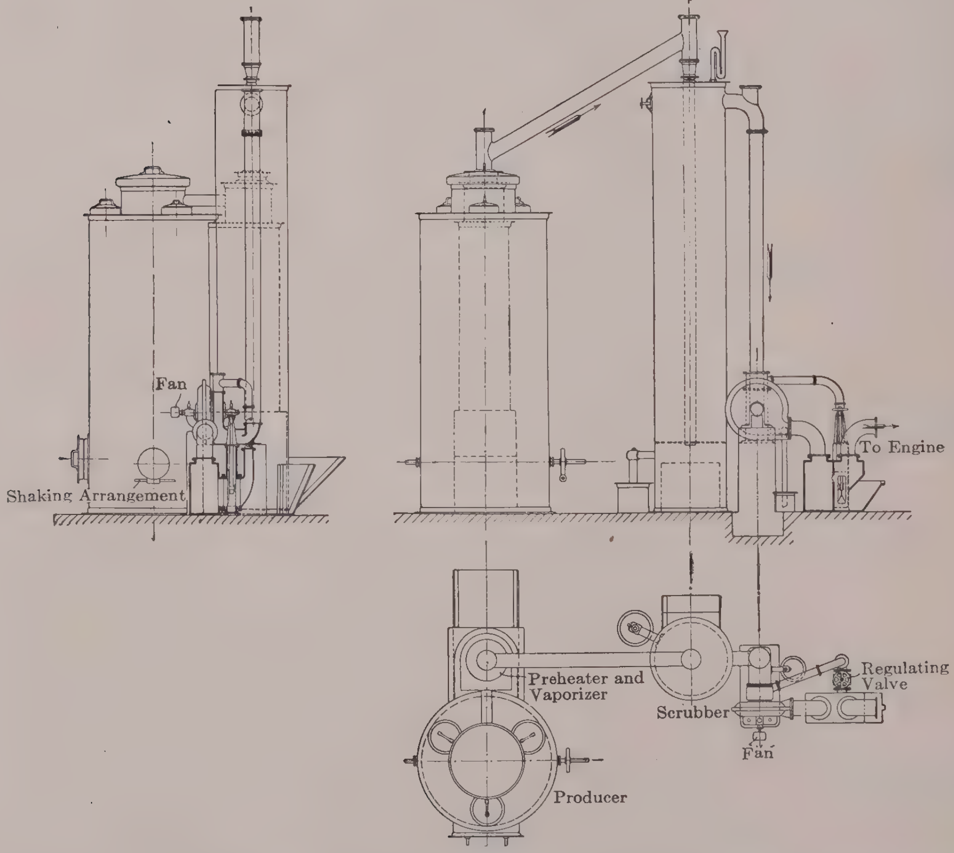


FIG. 434.—Deschamps Producer for Brown Coal and Peat.

In this producer all of the gas made is drawn out at the bottom. The producer proper, *a*, carries a fuel hopper or magazine *b*, in which is suspended a cast iron bell *c* whose lower end is located in the upper part of the incandescent zone. The gases of distillation gather in *c*, but they are drawn downward together with the pre-heated air, or air-steam mixture, coming from the pre-heater and vaporizer *e*, by means of a fan connected to the outlet *g*. In passing the gas heats *e*. The vaporizer *f* furnishes a certain amount of steam to the air as it enters the pre-heater coil, in case dry fuels are being gasified. Grate bars *d* may be shaken independently of each other and a poke bar inserted through the ash pit door serves to help keep the grate clean. For arrangement of entire plant, see Figs. 435–437 following.



FIGS. 435–437.—General Arrangement of a Producer Installation for Fuels forming Tar, Deschamps Type.

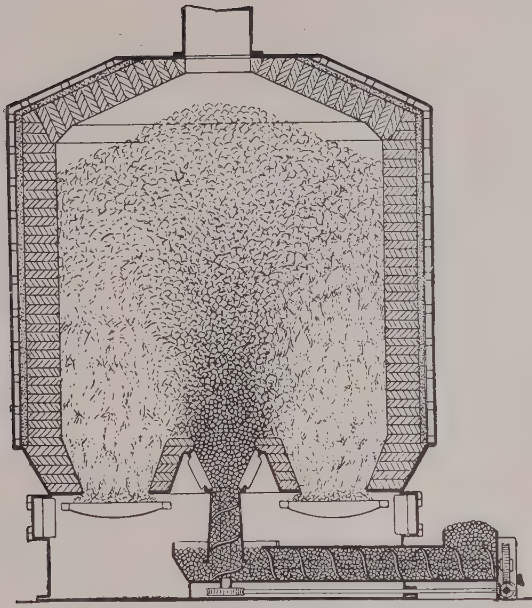
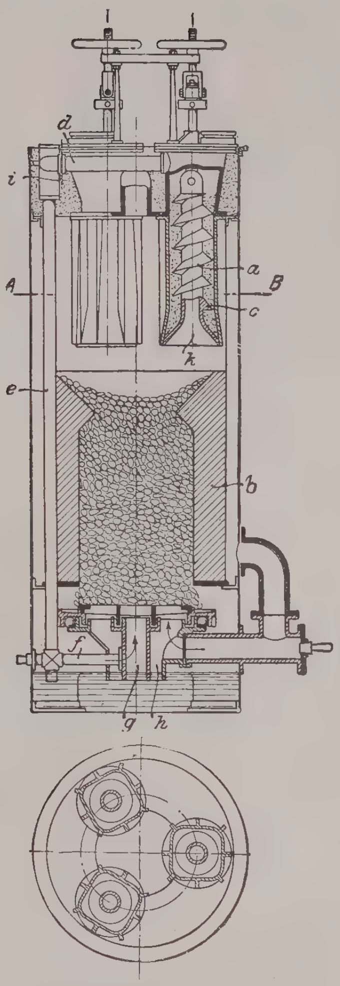


Fig. 438.—Double Producer, Emil Capitaine.

The fuel is forced in by a screw from below and between the two grates. The fresh fuel is coked and the gases of distillation, in passing through the incandescent zone, are either burned or fixed. The mechanical feeding of the coal is also intended to prevent coking or clinkering.

Details of Design and Dimensions of Gas Producers. The principal dimensions of producers, and especially of the combustion space proper, do not depend upon any theoretical considerations, but are based entirely upon experience and the results of practical tests. Fundamentally considered, the gas producer is a closed furnace in which a certain amount of coal must be burned on a given area of grate. How large the grate must be for that purpose depends, as in the case of other furnaces, directly upon the draft, the method of introducing the air, the kind of fuel, the depth of the fuel bed and what might be called the "load factor" of the installation. It is clear, however, at the outset that the grate and combustion chamber of a gas producer may be forced to much higher capacity than is possible in an ordinary furnace, because in the former the fuel is not completely burned but only gasified, and mostly with artificially increased draft.



Section A–B.

Figs. 439 and 440.—Brown Coal Gas Producer, Crossley Bros.

The drying and coking of the green coal takes place in the three retorts *a* which are located in the upper part *b* of the producer and are heated by the gas leaving. The gases of distillation, through *d*, *e*, *f*, and *g*, are led under the grate, which may be shaken by turning it about its axis. The air, which has been preheated, reaches the interior through the annular space *h*, as shown by the arrow. The retorts are separated from each other by slides *i*, and are charged and discharged in rotation. The coke is discharged by dropping the valve and screw *k* and then turning it about its axis.

Marine boilers with forced draft burn in the neighborhood of 30 lbs. of coal per sq.ft. of grate, locomotive boilers sometimes show as high as 100 lbs. per sq.ft. The former quantity of coal would in gas producers certainly yield $30 \times 55 = 1650$ cu.ft. of producer gas. Assuming that an engine uses 100 cu.ft. of this gas per B.H.P. hour, each sq.ft. of grate or cross-section of producer would supply 16.5 H.P., or conversely, each 100 H.P. would require in round numbers 6.0 sq.ft. of grate or producer cross-section. In practice similar figures are quite often found in smaller suction gas plants. With the ordinary sizes of anthracite coal, however, the combustion may be forced somewhat beyond this point, up to about 40 lbs. per sq.ft. of grate per hour, and larger installations are often operated at this figure.[1]

The average cross-section of grate or combustion chamber should therefore be

$$\text{for every normal H.P.} \begin{cases} 8.5 \text{ to } 7 \text{ sq.in. for installations up to 25 H.P.} \\ 7 \text{ to } 5.5 \text{ `` `` `` above 25 H.P.} \end{cases} \quad . \quad (1)$$

Very small fuels, as for instance buckwheat or rice coal, require a larger cross-section in order to obtain sufficient area for the passage of air and to be able to work with a thinner fuel bed. Fuels low in heating value also call for increased areas.

The internal height of the producer depends in the first place upon the heat consumption of the engine, the heating value of unit volume of the fuel, the time of combustion of one producer charge and finally also upon the behavior of the incandescent layers of fuel. Assuming that the heat consumed by the engine on the basis of fuel (that is, including generator losses) is 16,000 B.T.U. per B.H.P. hour, and that each charge of the producer lasts from 3 to 4 hours, the volume of the producer must be such as to store from 50,000–64,000 B.T.U. for every horse-power developed, or on the average 60,000 B.T.U. Now the heating values of some of the fuels based upon unit volume are about as follows:

---

[1] Translator's Note. These figures are evidently somewhat too high for American practice and they may be plainly labeled as applying to Continental practice only. The entire discussion following should be read with that fact in mind. As far as the writer's experience goes, gasification capacities on this side of the water range from 10 to 20 lbs. per sq.ft. of grate per hour, with the average perhaps at 15 lbs. He has seen only one manufacturer of gas producers claim more than that, up to 32 lbs. Actual data from tests in this country is as yet rather limited. Wyer in his Producer Gas and Gas Producers, says, p. 233:

"The rate of gasification in a gas producer is relative to the character of the coal used. The best rate determined by experience for a pressure type producer is 12 lbs. of coal per square feet of grate area per hour, although some makers have advised as high as 20 lbs. of coal. However, the exact limit of coal consumption is not known, as it is dependent upon a large number of empirical factors. Experience has also demonstrated that too rapid driving opens a wide door for the admission of adverse gasifying conditions."

As an example from English practice, Sexton, Producer Gas, p. 114, states that a Duff 8×10 producer will gasify about 1000 lbs. of coal per hour, giving a gasification of about 14 lbs. per sq.ft. of bottom area, and of course a larger amount per square foot of grate area. But even assuming that the grate occupies only 50% of the cross-section, this would mean a maximum gasification rate of 28 lbs.

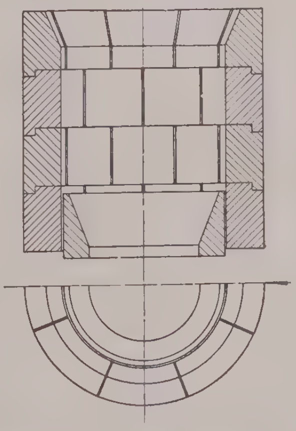
It should be noted that both of these examples are from pressure producer practice, while Güldner in the above speaks of suction-gas producers. The writer unfortunately has no data on hand for the latter type of producer. The kind of coal called anthracite by Güldner is not far different from our own hard coal, containing about 85% C and little ash.

1 cu.ft. = 50 lbs. of anthracite coal contains, deducting for losses, 565000 B.T.U.;
1 " = 25 lbs. of coke contains, deducting for losses, 340000 B.T.U.;
1 " = 50 lbs. of brown coal contains, deducting for losses, 340000 B.T.U.

Hence, for a charging period of from 3 to 4 hours, the generator volume should be

$$\left.\begin{array}{llllll}\text{for operation with} & \text{anthracite coal} & \text{at least} & .105 & \text{cu.ft.} \\ \text{''} \quad \text{''} \quad \text{''} & \text{coke} & \text{''} \quad \text{''} & .175 & \text{''} \\ \text{''} \quad \text{''} \quad \text{''} & \text{lignite} & \text{''} \quad \text{''} & .175 & \text{''}\end{array}\right\}\text{ for every D.H.P.} \quad . \quad (2)$$

The vertical height required for similar fuels increases with the size of such fuel. For very small sized fuel the fuel bed should be kept thin in order to reduce frictional resistance; the only way to get sufficient space in such a case is to increase the diameter of the producer correspondingly. On the other hand, a fuel bed that is too thin interferes with proper gasification, allowing both air and carbon dioxide to reach the upper generator space undecomposed, thus decreasing the volume of the gas and causing some of it to burn in the producer itself. It is better, therefore, to make the fuel bed too thick rather than too thin. This is of special importance for a coking coal, which, due to fusing together, may open up wide passages through the bed, thus seriously affecting the gasification process. Finally, an ample fuel space in the generator is one of the best assets where the fluctuations in the load are apt to be strong and long continued.



Figs. 441 and 442.—Sectional Producer Lining.

The construction of the generator shell and lining is the most important thing, determining the satisfactory operation of the producer as well as the value and life of the entire installation. In spite of this fact, far too little attention is paid to-day, even in their design, to the experiences gained in the building of industrial gas producers (Siemens type, etc.), and especially blast furnaces. There are suction-gas generators built in which the lining is constructed entirely at variance with the principles of gas technology, and the renewal or repair of which would call for the labor of skilled hands for days. Even the life of properly constructed linings is comparatively short; under the best conditions they may last about two years, under adverse circumstances often only a few months. Ease of repair and of renewal is therefore the thing to be provided for under any circumstances. Single-piece linings are not satisfactory even for small generators, for large producers they are absolutely unfit because they crack easily under heat and become useless, while their renewal is always accompanied by considerable expense.

The best way is to utilize in the first place a fire brick (not too fine grained or too hard burned on account of cracking) whose quality has already been tested in service, and to construct the lining of several rings joined as shown in Figs. 441 and 442. Each ring is divided into from 4 to 10 segments (or radial brick for the larger sizes). The lower ring wears out the fastest, and for that reason arrangements should be made to enable the renewal of this by itself with ease.

It is poor design to support the lining on a ring of metal solidly riveted to the generator shell. It is better to support it on a separate and divided ring of cast iron, whose various parts may be easily replaced through the ash-pit doors when the side next the fire has been burned out or melted away. (See Fig. 426.)

The flat grate so generally used in producers for the medium sizes of anthracite and for coke has air spaces varying in width from $\frac{3}{16}''$ to $\frac{7}{16}''$. Too wide air spaces allow too much of the fuel to fall through, while those too narrow soon clog up with ash and clinker, thus interfering with the cleaning of the grate. Very small-sized fuel requires the use of step or basket grates; while some other fuels, as yet little used, also call for special grates. Still others, like wood and charcoal, may be gasified in special generators without grates.

As determined from practical experience, the starting fan in installations up to 50 H.P. should have a capacity of about 175 cu.ft. of free air per minute. Larger installations require capacities of from 350 to 700 cu.ft. per minute, depending upon horse power. On account of low cost it is usual to employ hand-operated fans in which hand-crank and multiplying gear form a part of the outfit. It is better, however, in every case, and in large installations indispensible, to have the fan operated by a special source of power (small water wheel, electric motor, etc.) in order to leave the attendant free for his other duties during the period of heating up.

2. Gas Washers and Purifiers. The gas as it leaves the producer carries along with it a certain amount of fine dust which must next be removed. Depending upon the kind of coal used, the gas also contains more or less chemical impurities, such as tar, sulphur vapor etc., each of which may cause trouble in operation, and should therefore be removed as far as possible. The common types of wet scrubbers are, if properly constructed, usually able to successfully throw down most of the mechanical impurities in the power gas. In spite of this washing, however, there is usually still enough fine dust left in the gas to cause scoring of the internal rubbing surfaces of the engine. As a matter of safety, therefore, it is *always* best to pass the gas through a dry purifier after it passes the scrubber.

The chemical impurities, when they occur in any considerable quantity, cannot be successfully eliminated by any of the common washing apparatus. Neither hard coal nor coke cause any trouble in this respect, because the quantity of tar found in the gas is so small that it can be easily handled with the simple means in common use. The latter, however, completely fail with gas from bituminous coal, for which reason alone the tar-forming constituents of this gas should be changed to fixed gases in the producer itself, i.e., they should not be allowed to reach the cleaning apparatus. An exception to this rule is found in some of the very large industrial gas-generating plants, in which special apparatus is used to separate the tar and the other impurities and to obtain them as by-products.

Wet purifiers (washers, scrubbers) usually consist of tall cylindrical vessels through which the gas passes from bottom to top, being washed by means of fine streams of water which it meets in its ascent. In order to promote a thorough contact between gas and water, the purifiers are usually filled with coarse coke, sometimes also with wooden baffles, etc. There is a free space at the top to allow the water to spread evenly, and one at the bottom to serve as a settling chamber for water and mud. The chamber should be provided with ample exit and cleaning openings.

The water-swept space of the scrubber should be at least .7 cu.ft., better still from .9 to 1.1 cu.ft., for every engine horse-power. The height should be as great as convenient, in order to provide long paths for water and gas. Purifiers of low height but large diameter should be used only when the water is very thoroughly sprayed and distributed.

The water may be finely divided by any of the well-known spray apparatus or by baffle plates. Regarding the use of the former, it should be noted that fine openings may be stopped up internally by impurities in the water and externally by tarry deposits, in either case causing trouble. For that reason any spray apparatus used should be so arranged that it can be easily taken out and cleaned. The dirty water should be taken out at the bottom of the washer by a pipe of ample size into a syphon or seal box. The latter should be tightly closed at the top to confine the very obnoxious odor of the water, and further should be located at least 12″ lower than the scrubber to prevent a sucking back of the water.

The so-called *centrifugal washers*, if properly made, both clean and cool the gas very thoroughly. The simplest form of this washer consists of a fan (exhaust fan or blower) in which water is sprayed into the casing (see Figs. 432 and 435). There are, however, special centrifugal washers[1] on the market. (See Fig. 431.) The latter are very well adapted to large installations and have been used for some time up to the very largest sizes (in blast-furnace gas installations for example). The thing that prevents their general adoption in commercial gas producer installations is the fact that they require a power drive, which, in most cases, is not available in the producer room.

Dry purifiers are cylindrical or box-like vessels which are furnished with from 2 to 4 horizontal perforated partitions which serve to support the purifying material. The latter may consist of wood shavings or sawdust, excelsior, chips of iron or steel, slag wool, etc. Any material that contains fine, sharp particles, or which disintegrates under the action of the gas and thereby forms fine and hard particles which the gas may pick up, is unfit for use. Slag wool and iron chips may be specially harmful in this direction. The free passage for the gas should be the greater the finer the purifying agent; sawdust, for that reason, requires the most roomy dry purifiers. On the average, the approximate volume of dry purifiers should be about .2 cu.ft. per engine horse-power; but, where possible, from .3 to .5 cu.ft. should be allowed.

Several firms also connect into the gas mains just beyond the scrubbers a tar separator or extractor. These are generally based on the principle of Pelouze, according to which tar vapor when made to strike metallic plates at right angles under high velocity, will condense and gather on these plates in drops. See Fig. 429. Care must be taken to see that these baffle walls can be exchanged quickly after having been coated with tar. The best-known, but at the same time least desirable, form of this apparatus consists of a cylindrical wire brush, which is very often placed in the gas main close to the engine. These brushes, even when clean, increase the friction resistance in the main considerably, and becoming dirty after only a few days of operation, so seriously interfere with the suction that the normal load of the engine can no longer be attained.

The path for the dirty gas, that is, the piping between producer and the first scrubber, should be as short as possible and so arranged as to offer the least chance for the lodgment of any soot or ashes. Whenever these are deposited, however, ample opportunity for cleaning in the simplest and quickest way should be provided. This requirement is hard to fulfil if, as is usual, the vaporizer or pre-heater is built into the gas main between generator and scrubber. It is equally out of place to put the dry purifier close to the engine, in which case the still partially uncleaned gas is forced to pass through considerable lengths of pipe, giving it a chance to clog them up. All cleaning apparatus belongs close to the producer and should not be placed in close proximity to the engine.

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[1] For example, that made by Edward Theissen, Munich. See footnote, p. 282.

The *gas receivers* often used between the cleaning apparatus and the engine are of little value for purifying the gas. They may act as gas holders or pressure regulators only when their volume is made much greater than is usually the case. This volume, if the receiver is to serve any purpose at all, should never be less than ten times the piston displacement.

It should not be forgotten to supply the pipe line just in front of the engine with a suitable arrangement through which the system can be cleared of air and poor gas. The *purge pipe* should not be combined in any way with the try-cock, used to try the quality of the gas, but should be a line leading directly outside to avoid fouling the air in the producer room. The try-cock should be opened only after the poor gas has been cleared out.

Last, but not least, the designer should, besides taking into account technical and practical requirements, also heed whatever government or insurance specifications may exist in his locality regarding the construction and erection of gas producer installations. Such supervision does not as yet exist in many places, but the designer will do well to take the most strict of those in in existence, as a guide to avoid encountering difficulty in the sale of the apparatus in any territory.

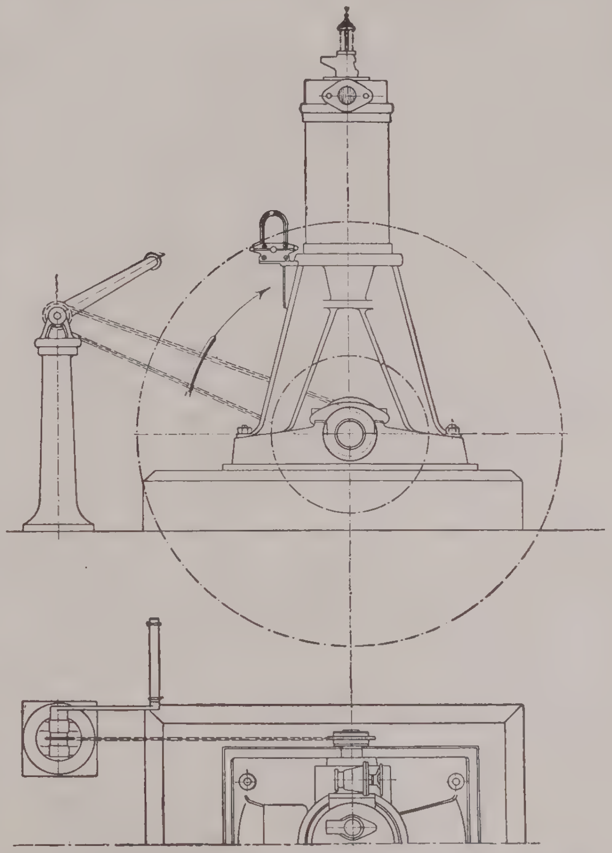
II. Starting Apparatus

An internal-combustion engine is not self-starting, but must by some suitable external means be set in sufficiently rapid motion before it can do the work required to overcome its internal resistance. If the starting is done by "cranking" or turning the shaft, the starting agent may continue its action even after the first ignitions have taken place, in which case both sources of power combine to start the engine. If, on the other hand, the source of power for starting is made to act in the cylinder, it is usual to impart to the flywheel only sufficient velocity so that the inertia of the mass may be enough to overcome the starting resistance. After this velocity is attained, the starting agent is shut off, the first ignition takes place in the cylinder and soon brings the engine up to normal speed. To reduce the starting resistance it is usual to reduce the compression on starting as far as the kind of fuel used will allow, for rich fuels to from 1–2 atm., for lean fuels to from 3–4 atm. This is done by forcing a part of the charge through the exhaust valve, i.e., not closing the latter until some time into the compression stroke. It is also common, at least in large engines, to so retard the spark that ignition cannot occur until after the dead center is passed in order to avoid the back-firing so dangerous to attendants.

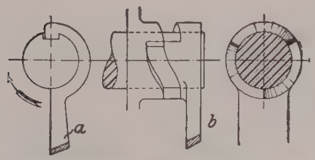
The usual methods of starting serve their purpose only when the engine is light, but will not start an engine under load. Only twin-cylinder engines may be started, at least under partial load, when one cylinder is operated with compressed air while the other operates on fuel. But this scheme requires large air storage-tanks and is even then not quite certain.

A suitable starting gear should in all cases be a part of an up-to-date engine installation. For large engines it is of course a necessity, but even for the smaller units it is a good thing from the standpoint of safety. The starting of an engine by turning the wheel is dangerous for unskilled hands, and German trade-unions, at least of late, therefore require that even small machines shall be furnished with proper starting apparatus.

1. Hand Cranks. It is possible to start engines up to 10 H.P. by means of hand cranks. Larger engines, up to say 20 H.P., may be started in a similar way if the motion is transmitted through gears or sprockets and chain, as shown in Figs. 443 and 444. In such a case it is best to arrange the drive so that the operator is able to bring his greatest power to bear on the crank at the moment of greatest resistance in the engine, i.e., near the end of the compression stroke. In four-cycle engines this means that the transmission ratio should be either 2:1 or 4:1.


Figs. 443 and 444.—Starting apparatus with safety arrangement against back-firing, 15 H.P. Güldner Engine.

The types of hand cranks *a* and *b*, shown in Figs. 445–447, automatically lose their grip on the shaft as soon as the first normal explosion occurs, but they remain in mesh if through back-firing or pre-ignition the shaft should start in the opposite direction. This may, under the circumstances, be dangerous to the attendants and may be avoided by the use of safety cranks which free themselves from the shaft when it starts in either direction of rotation. An example of this starting crank is that made by Dr. W. Heffter of Berlin, N.W., in which, when the shaft starts in the normal direction, an axial bolt simply unscrews out of the end of the shaft, while for rotation in the opposite direction the connection is broken by a ratchet arrangement. This safety crank is made for engines up to 16 H.P.; for larger sizes gear or chain drives are used in connection with it.



Figs. 445–447.—Starting Cranks.

2. Mechanical Starting Apparatus. Where a second engine is available, it, or the transmission it operates, may be used to start the engine. Special arrangements for this purpose are usually not necessary. The scheme formerly used, of supplying each larger machine with its own starting engine, is seldom employed to-day. Where water under high pressure can be had, the use of a hydraulic piston motor of some kind might be considered, especially if the water used can afterwards find application in the jackets of the main engine.

3. Starting by Means of the Fuel Mixture. The general idea of this method is to place the engine crank 10 or 15% beyond head dead center, to fill the combustion chamber with the fuel mixture, and to cause ignition by snapping the spark by hand. The piston thus receives an impulse which is usually sufficient to impart enough energy to the flywheel to last for several further turns, during which the normal charge is drawn in, compressed and ignited. There are several methods of carrying out this scheme, which differ among themselves in the mechanical means employed. Clerk caused the fuel pump of his 2-cycle engine to pump the normal fuel mixture into a reservoir during the time that the igniter was shut off and the engine slowing down. The compressed mixture was first led into the power and pump cylinders and acted as the direct source of power to cause the first rotation. At the end of expansion the mixture was not allowed to escape, but was compressed on the next instroke and ignited. Lanchester utilizes the sucking action of the exhaust pipe to draw a combustible mixture into a vessel connected to the combustion chamber during the time that the exhaust valve was open. This mixture is at the desired time exploded by means of an open flame in connection with the above-mentioned vessel. The gases of combustion immediately flow over into the combustion chamber and give an impulse to the crank which has previously been set at the proper starting position. This arrangement was afterward improved by connecting gas and ignition cocks directly to the cylinder. The latter, during the slowing down of the engine, has been thoroughly scavenged and filled with clean air. On starting, the gas cock at the side of the cylinder is opened. A second cock on top of the cylinder allows of the escape first of some of the air displaced by the gas, and soon thereafter of the mixture formed. The latter is ignited by an open flame in front of the cock. As soon as the needle flame formed shows that the composition of the mixture is right, the gas cock at the side is closed. This stops the flow of mixture out through the ignition cock, the flame there burning strikes back into the interior of the cylinder and explodes the mixture still remaining. The pressure so generated closes a small check valve in the ignition cock and gives an impulse to the piston. The smaller Simplex (Cockerill) engines are started in a similar way. The large blast furnace gas engines of the same firm are put in operation by pumping a gasoline fuel mixture behind the piston when in proper position, compressing this mixture slightly and exploding it by means of electric spark. The scheme used by the Berlin-Anhaltische Maschinenbau Gesellschaft for the starting of their large engines, consists in allowing a small gas engine to compress a part of its fuel mixture through a third valve into the combustion chamber of the large machines. During this operation the piston of the latter is held in the starting position by blocking the fly-wheel. The mixture is exploded by electric spark, the impulse breaks the cast-iron block at the wheel, freeing the shaft.

Starting by mixture is a simple process in the case of illuminating-gas engines. For other fuels it is somewhat complicated and not absolutely reliable. If the engine should fail to start at the first trial, which may occur even with the best of machines, there is often not enough mixture in reserve for a second attempt, and it becomes necessary to go through all of the operation again, which often takes considerable time. Especially in the case of large engines, the work of refilling the reservoir or cylinder with mixture is generally troublesome and tedious.

4. Starting by Compressed Air. This method, which enjoys general preference to-day, was already in use in the old constant pressure 2-cycle engines of Brayton and Simon, in which air in receivers under a pressure of from 60–75 lbs. was constantly available. The pressure of the scavenging air used in our modern 2-cycle explosion engines is not high enough for starting. The starting air is usually compressed either in the main cylinder or by means of an independent small compressor. According to the former scheme the piston of the main engine during the slowing down period, after the fuel is shut off, compresses the air drawn in into a special receiver through a valve, which is generally used also as the starting valve. The receiver should be of such size that the engine can compress the air up to 75 or 90 lbs. With this pressure it becomes necessary to give the piston several impulses on starting. On the other hand, where an independent compressor is used, it is advisable to compress the starting air up to from 150 to 225 lbs., and to accomplish the starting with few but powerful impulses. With an installation properly arranged and adjusted (good gas, proper mixture, correct spark timing, right position of piston, etc.), a single impulse is generally sufficient to start with certainty engines, with ordinary fly-wheels, up to 150 H.P. Especially where the starting valve is operated by hand, it is of importance to start with one or at most two impulses, because the proper handling of the valve becomes difficult as soon as the engine speeds up.

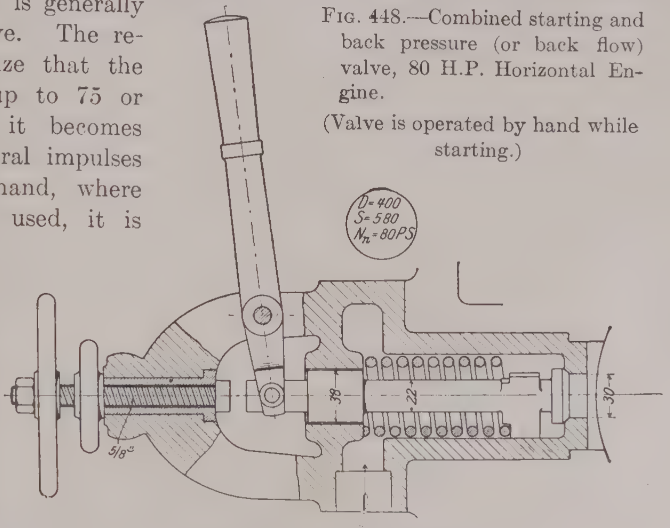


Fig. 448.—Combined starting and back pressure (or back flow) valve, 80 H.P. Horizontal Engine.

(Valve is operated by hand while starting.)

If the valve through which the main piston delivers air into the receiver is also used as the starting valve, its valve disk must open outward, and must, during normal operation, be held against its seat by means of a screw, see Fig. 448. Such a valve, however, will leak easier than if the disk opened

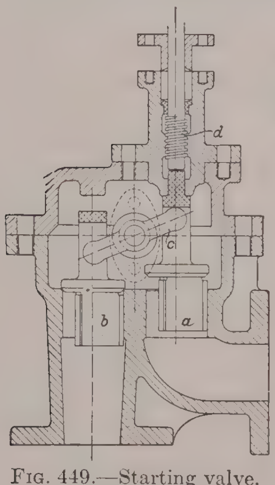


Fig. 449.—Starting valve, Hornsby & Sons.

(For location of this valve, see Figs. 240–242, p. 197. Fig. 450 shows the method of operating it.)

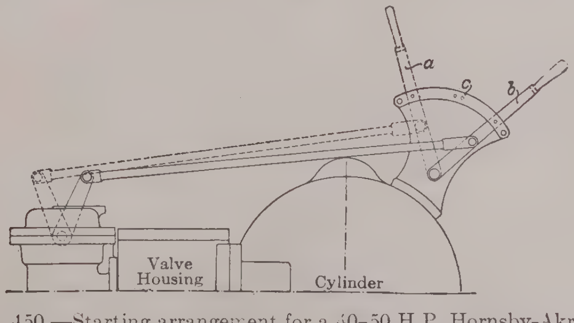
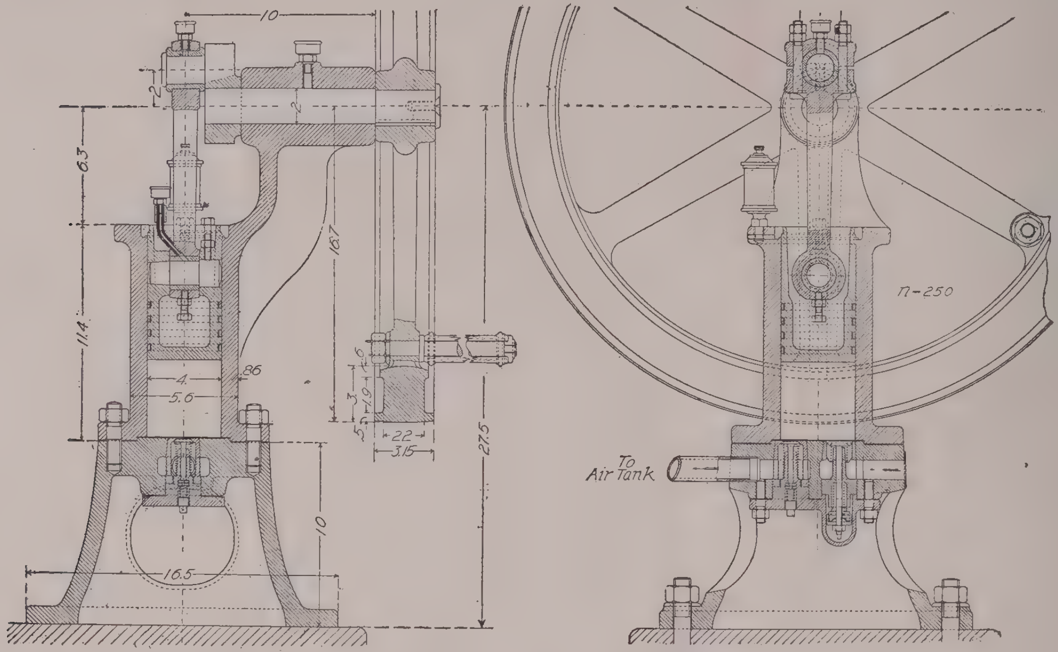


Fig. 450.—Starting arrangement for a 40–50 H.P. Hornsby-Akroyd Oil Engine.

[Operated by hand so that air is admitted at the beginning of the third stroke (expansion stroke)].

For starting, the lever is placed in the position *a*, Fig. 450. Air compressed to 75 or 90 lbs. then flows through the valves *b* and *a*, Fig 449, into the cylinder. The air is obtained by letting the engine act as a compressor when slowing down for a stop. For this the lever is put in the position *b*, which raises valve *b* while *a* acts as a back pressure valve. In the middle position *c* of the lever, Fig. 450, the internal lever *c*, Fig. 449, leaves both valves free. *a* is then held down on its seat by *d*, which is the normal operating position.

inward and were held against its seat by the force of the explosion. A further disadvantage of the "slowing down" method of obtaining compressed air for starting consists in the trouble of getting air for starting the first time and the refilling of the receivers when the charge of air is suddenly lost through leaks.



FIGS. 451 and 452.—Starting Air Compressor for Engines from 60–80 H P.

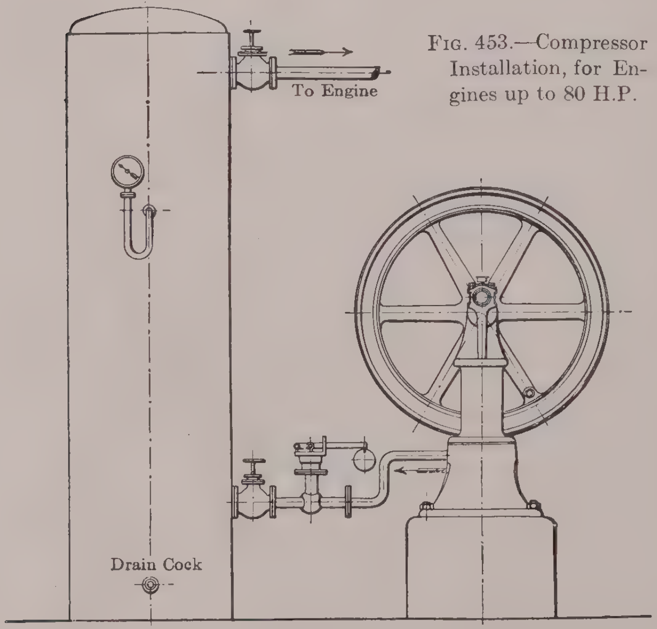


FIG. 453.—Compressor Installation, for Engines up to 80 H.P.

For large engine installations preference should be given to independent air compressors driven by their own source of power wherever possible. Any connection or interdependence of air pump and main engine is bad, because in case of repeated failure to start, the air will be used up without any chance of renewal by means of the engine. This may lead to serious trouble. Under such circumstances, recourse has already been had to compressed carbonic-acid gas, which can be bought in steel flasks in the market. But this scheme is not entirely sure, since the residual carbon dioxide makes the mixture poor and hard to ignite. This latter fact has probably caused the Gasmotorenfabrik Deutz to abandon its patented method of starting with compressed exhaust gas. (This method consisted in allowing a small part of the burned gases at each explosion to flow through a small heavily loaded valve into a receiver from which they were afterward used to start the engine. The valve mentioned was entirely closed when the receiver pressure indicated from 100 to 120 lbs.)

An example of a small starting compressor is shown in Figs. 451 and 452. This is designed to compress air up to 180–225 lbs., and requires from 1.5 to 2.0 H.P. to operate it. Since the compressor is only used a few moments every day, the cooling of the compressor cylinder may be dispensed with. The machine is normally operated by belt, but the wheel is also fitted with a crank, which may be used as a makeshift. The arrangement of a compressor of this type with its receiver may be seen from Fig. 453. The volume of the tank should not be less than .35 cu.ft. for each normal engine horse-power, i.e., about 10 times the piston displacement. In important installations the use of two such tanks is advisable, both being kept under pressure and the second used as a reserve for the first.

5. Starting by Electricity. Gas engines which are used to drive electric generators may be started, in case a storage battery or another source of current is available, by operating the dynamo for a few turns as a motor. The use of current in this manner, of course, assumes that there is sufficient capacity in the battery. The actual arrangement of the system of wiring used for the purpose may be seen from the diagram of connections, Fig. 454. This shows that the normal wiring system is increased only by the addition of a double-throw switch and a connection shown in broken line in the figure. The method of operation is very simple, it is merely necessary to close a double-throw switch, to regulate the position of the charging or discharging levers and to adjust a resistance box. As soon as the engine picks up its cycle and the starting current drops to a certain point, an automatic circuit breaker acts, and the dynamo, which up to this point has acted as a motor, operates under no load until the proper switches are thrown and the circuit is closed.

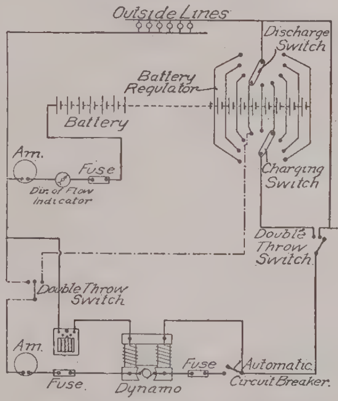


Fig. 454.—Lay-out of Connections for using Generator as a Starting Motor.

The use of a storage battery as the source of starting current, is only feasible for the smaller engines, say up to 50 H.P., and even for these the service is rather severe on the battery. It is quite possible to totally destroy small batteries in a few days when used for starting, especially if the matter is not skilfully handled. The engine should always be started with the piston in position at the end of the compression stroke, in order that the electric motor during the next three strokes is required to accelerate only the fly-wheel and to have the kinetic energy of the latter aid in the next compression stroke. It is totally wrong, even with the largest and most powerful battery, to commence starting with the piston, for instance, at the end of the suction stroke, because in such case the motor must not only accelerate the moving parts, but in addition is called upon to overcome the maximum resistance at the crank. It is probably best in all cases to start by holding open the exhaust valve, and thus calling upon the electric motor only for enough energy to bring the fly-wheel up to the desired speed, and after this is accomplished to set the valve gear to normal position.

The figures in the following table will give some idea of the power consumed in overcoming the starting resistance:

TABLE 27

POWER CONSUMPTION OF STARTING-MOTORS

40 H.P. Deutz engine driving A. E. G. Dynamo, 110 V., 200 Amp.				60 H.P. Deutz engine driving A. E. G. Dynamo, 110 V., 450 Amp.			
1. Second	112 volts	30 amp.		1. Second	110 volts	50 amp.	
2. "	111 "	40 "		2. "	110 "	90 "	
4. "	111 "	45 "		3. "	110 "	100 "	
6. "	110 "	60 "		4. "	110 "	120 "	
7. "	110 "	90 "		5. "	110 "	120 "	1st ignition engine starts
8. "	110 "	110 "	1st ignition engine starts	6. "	110 "	100 "	
				7. "	110 "	90 "	

According to an empirical formula, based by Leroy upon the results of several tests on the subject, the maximum power required to start by electricity in the manner described is approximately $\frac{2}{5}$ of the maximum capacity of the dynamo concerned.[1]

III. Mufflers

The noise caused by the air drawn into the engine and that due to the exhaust is annoying even in small engines; in large machines it may become a public nuisance. The various kinds of mufflers used to diminish or abolish this noise show a number of different constructions, but they all depend for their action either upon the gradual decrease of the air or gas velocity, or upon the principle of the Helmholtz "resonator," in which the inlet or outlet pipes are enlarged into one or more equalizing chambers into which the sound waves expand with consequent reduction of intensity. The latter method is the one mostly used for exhaust mufflers.

1. Inlet Mufflers. Besides decreasing the noise, inlet mufflers also serve the purpose of freeing the air from the coarser mechanical impurities and of any considerable amount of water that it may carry. Simple vessels, or receivers, used for this purpose are usually given a volume at least equal to five times the piston displacement, and where possible more. Baffle plates, perforated partitions, and similar means for reducing the noise make it possible to make the muffler volume considerably smaller. The wall thickness of the mufflers depends directly upon considerations of manufacture; the material mostly used, that is, cast iron, requires a minimum thickness of from $\frac{1}{2}$ to $\frac{5}{8}''$. Sheet-steel mufflers are most often used in combination with baffle plates, partitions, etc.; without these such mufflers require considerable wall thickness in order to prevent resonant vibrations in the muffler walls themselves.

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[1] Revue industrielle, 1898, p. 229, according to L'industrie électrique, 1898.

It is possible, in many cases, to sufficiently deaden the noise by taking the air from the base of the engine, or by shaping the end of the pipe so that the inlet cross-section is divided into a number of narrow slots (.04 to .08″) of ample length. In the former case, the air inlet to the interior of the frame or bed should not be placed in the side next to the fly-wheel because of the greater quantity of dust there stirred up, which latter then easily reaches the cylinder.

It is distinctly not advantageous, in the case of vertical engines, to draw the air from a closed crank case. Under such conditions the air is considerably pre-heated, is consequently less dense, and may further be contaminated by oil vapors. On the other hand, taking the air from the room itself has the advantage of aiding in the ventilation and further may serve to draw out of the room some of the oil vapor formed, a fact more or less pleasantly noticeable, especially in small installations.

In the case of large engines the complete abolishment of the inlet noise is possible only at considerable expense, and then only with means that in most cases require considerable room. The most common way of doing this is to construct special pits or underground suction chambers which are connected to the engine by means of masonry conduits and have their own inlet openings in the open air, where there is least dust. In some cases large engines have been allowed to draw air from interior chambers in their own foundations, but the scheme is permissible only when the chambers have been finished inside to guarantee absence of dust and have otherwise been so constructed that a transfer of the sound waves cannot take place.

2. Exhaust Mufflers. Exhaust mufflers, in order to be fairly efficient as regards deadening of noise, require a volume equal to from 15 to 20 times the piston displacement. In general, however, the volume is made only from 6 to 8 times the piston displacement at the expense of efficiency. To fulfil the strictest requirements as regards muffling, it is not only better but generally also cheaper to connect two or three smaller mufflers in series. As in the case of inlet mufflers, the addition of any scheme to gradually increase the volume or to gradually change direction of flow by baffles, etc., enables the use of exhaust mufflers of smaller volume, but in the use of such means care should be taken to see that the back pressure on the engine is not too greatly increased. In general, the slower the flow of gas from the cylinder, i.e., the slower the opening of the exhaust valves or ports, the smaller may the muffler be made. For that reason, engines with a ring of exhaust ports, for instance, require larger mufflers than machines having exhaust valves. In multicylinder engines it is a good scheme to furnish each cylinder with a muffler, in order to prevent any possible interference of the outflowing gases.

Cast-iron muffle pots having an internal diameter of D inches may be given a wall thickness of about

$$S=\frac{D}{50}+.25'', \text{ but } S \text{ should never be less than } \tfrac{5}{8}''. \quad \ldots\ldots \quad (1)$$

Their height or length should be about

$$H=1.25 \text{ to } 1.75\,D. \quad \ldots\ldots \quad (2)$$

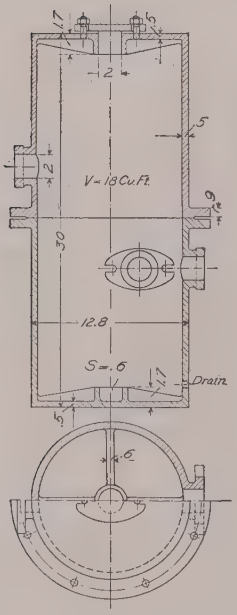
Above D=about 40″, the use of a wall thickness according to Eq. (1) results in cumbersome and heavy constructions. It is consequently better above this size to use sheet steel, as, for instance, old boiler plate from $\frac{3}{8}$ to $\frac{5}{8}$″ thick.

The inlet and outlet openings of these muffle pots should be separated from each other as far as possible, the one for instance should be placed diametrically or tangentially above the bottom, the other axially in the top or head, see Figs. 457–458. Just above the bottom there should be a drain-opening at least 1″ in diameter.

The sheet-steel mufflers bought in the market rarely form a completely successful substitute for the muffle pot. They are internally strongly corroded by the exhaust gases even when galvanized, and in most cases increase the back pressure. Further, if the steel is not of ample thickness and the surfaces not sufficiently stayed, each exhaust may cause noisy vibrations in the muffler itself.

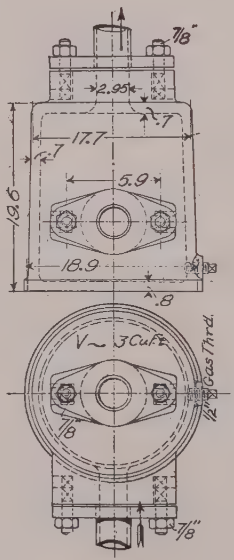
Large muffle pots sometimes cause considerable annoyance on account of radiation of heat. For this reason, in several of the large engines, a part of the cylinder-cooling water is sprayed into the muffler, or the exhaust line may be used as the exit for all of the jacket water. In such cases, however, the greatest care should be exercised to see that the water can, under no circumstances, flow back into the cylinder or be drawn back by the piston. From a practical standpoint this scheme of cooling the exhaust gases has the disadvantage that the pipe is very apt to rust through and must be very thoroughly drained in cold weather. Further, the exhaust is constantly accompanied by a cloud of steam, which makes impossible a visual judgment of the quality of the exhaust gases.

Designs of Mufflers:



Figs. 455 and 456.

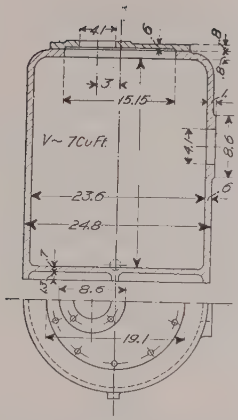
Exhaust Muffler, 6 H.P. Engine.



Figs. 457 and 458.

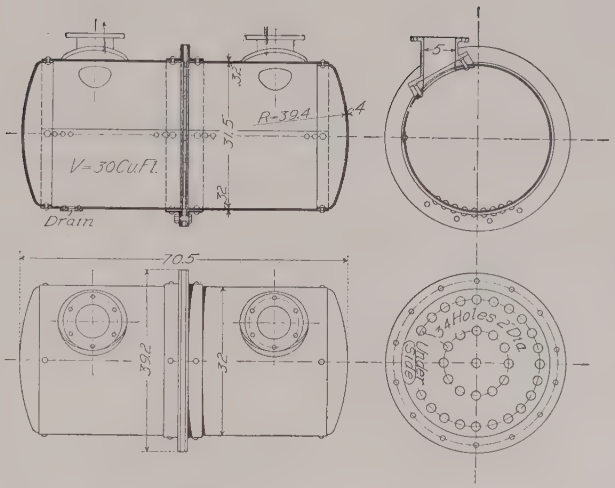
Muffler for a 12 H.P. Hornsby Engine.

The quieting of the exhaust noise in large gas engines offers even greater constructive difficulties than the muffling of the inlet. The common muffle pots finally assume very large dimensions, and even then do not fully serve their purpose. The best scheme is in this case, also, the construction of underground masonry ducts into which the gases are allowed to enter after the greater part of the expansion has been completed in an iron muffle pot. Probably the most complete deadening of the noise is obtained if, after the exhaust gas has been allowed to expand into the ordinary muffler or into a duct, it is drawn out of the latter by means of an exhauster running in a water spray, and thus discharged into the open air in a steady stream. The method, however, is rather costly, since it requires a steady source of power, a constant supply of cooling water and is subject to frequent repairs.



Figs. 459 and 460.—Exhaust Muffle for a 25 H.P. Engine.

(The cover may be shifted so as to bring the flange opening for the outlet pipe in the desired position.)



Figs. 461–464.—Steel Plate Exhaust Muffler, 80–100 H.P. Engine.

IV. Cooling Arrangements

Although the smaller sizes of automobile engines are often cooled by some method of air cooling, stationary engines are almost without exception cooled by water. The cooling at least extends to the combustion chamber and that part of the cylinder uncovered by the piston in its travel, but it is in many cases also extended to the other less highly heated parts.

Experience shows that on the average one-third of the total heat supplied per effective horse-power hour is lost in the external cooling medium. With the liberal assumption that the consumption of heat per H.P. hour is 12 000 B.T.U., it will be seen that the cooling medium must be able to carry off 4000 B.T.U. per H.P. hour. With a temperature range of 90° (from 60 to 150°), this corresponds to a consumption of cooling water equal to 45 lbs. (5.5 gals.) per H.P. hour, assuming that the hot water is wasted.

In the larger engines the heat consumption per H.P. is considerably smaller than above assumed. This has the effect of reducing the amount of cooling water required (to about 4.25 gals.) especially since the larger the machine the smaller will be the proportionate heat loss to the jacket. On the other hand, certain features of construction and of operation of large engines make it usually necessary not to heat the jacket water above about 125°, which will again tend to increase the quantity used. In case the fuel mixture carries easily inflammable gases, like hydrogen, it is well, in order to be on the safe side, to estimate the amount of cooling water required per H.P.-hour at not less than from 5.5 to 8.0 gallons, based upon maximum capacity.

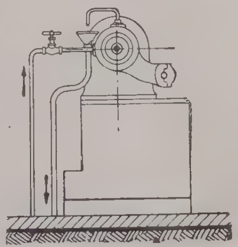


Fig. 465.—Direct Cooling of a Horizontal Engine.

It is of advantage to construct separate inlet and outlet lines for the jacket water to the most important engine parts requiring cooling (cylinder head and barrel, valve cages and stuffing boxes, exhaust valve, piston and piston rods, etc.), so that the supply to each can be independently regulated. For the water-cooled pistons of large engines a separate line is indispensible since the water must be under a pressure of from 60 to 75 lbs. per sq.in., while for the other machine parts a head of from 25 to 35 feet is usually quite sufficient.

Wherever it is not possible to obtain sufficient cooling water, that is, where it is obtained only with difficulty, or where its cost is high, it becomes necessary to cool the hot water by some artificial means in order to use it over and over. In this way only the escaping vapor or steam must be replaced and the consumption of cooling water may be cut down .8 gallon per B.H.P. hour. The cooling arrangements for this purpose in small installations usually consist simply of tanks whose capacity or superficial surface is sufficient to radiate the heat taken up from the engine during a limited period of operation.

A somewhat forced means sometimes used to reduce the consumption of cooling water, consists in not only heating the water in the cylinder jacket, but in actually boiling it in the jacket space. Thus each unit of water carries away not only the heat of the liquid but in addition also the latent heat of vaporization, which is approximately five times as great as the former. Several disadvantages connected with this scheme, however, make it generally undesirable for use in stationary engines. In small alcohol engines it may under certain circumstances give satisfactory service. Altman in Berlin was the first to use this method of cooling in his alcohol locomobiles, but the idea is common property and had already been used by Lenoir in 1860 in his first gas engines.[1]

To cool the jacket water, there may be used

1. Cooling Tanks, Fig. 466. They may be used in installations up to 50 H.P., and are constructed of galvanized sheet steel $\frac{1}{16}$ or $\frac{3}{32}''$ thick. The bottom is either made of somewhat thicker stock or suitably stiffened. The height may be from 1.75 to 2.5 times the diameter. The capacity may be made approximately equal to

$$V = 1.4\, z\, N_n \text{ cu.ft.} \quad . \quad . \quad . \quad . \quad (3)$$

provided the engine, of N_n B.H.P. normal capacity, is in operation z hours daily ($z \underset{>}{=} 10$). It is recommended, however, to use two or more tanks if the necessary capacity V exceeds 175 cu.ft. The flange for the outlet pipe is placed either in the bottom or in the side close to the bottom while the inlet opening is situated from 6 to 10″ from the top. To best fulfil its purpose the tank should be placed in a cool, airy spot, but of course not too far from the engine. On account of the danger of freezing it is not safe to place it outdoors.

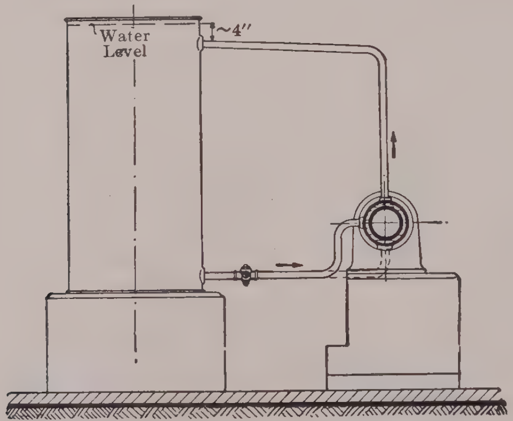


Fig. 466.—Cooling System for Horizontal Engine, Natural Circulation.

Generally the difference between the temperatures or weights of the two water columns is enough to maintain a slow, but under favorable circumstances sufficiently rapid circulation between engine cylinder and tank. Where the pipe lines are long or have elbows, and generally in large installations this method of producing circulation is not sufficiently dependable, and it is better to cause circulation mechanically by means of a rotary pump. The periodic addition of fresh water to the system to replace that lost by vaporization is most easily made automatic by the use of a float-regulating valve.

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[1] See Dingler's Polyt. Journal, 1861, p. 2.

2. Extended Surface Radiators. These are connected by short pipe lines, top and bottom, to the jacket space, and give good service in small installations if they can be erected in an airy place. The internal width of these radiators is from 3 to 4″, and the usual allowance is from 30 to 40 sq.ft. of superficial surface per B.H.P. per day of 10 hours.

3. Cooling Towers. These use either natural draft or fan draft to cool the hot jacket water. Cooling towers with natural draft are used in large gas-engine installations only when absolutely necessary, on account of high first cost and large space required. For every 1000 lbs. of water cooled, they require, when 12 to 14 layers of plank or crib work are used, from 5 to 10 sq.ft. of floor space.

Fan coolers are quite widely used for portable oil engines. The hot water enters these coolers at the top through a spray nozzle, flows down over 10 to 15 partitions or baffle plates, around which air is blown from below, and gathers in a reservoir at the bottom, cooler by from 30 to 45°. From here it flows to the cylinder and back to the spray nozzle. Oil locomobiles powered with 8 to 12 H.P. use fan blowers having from 12 to 16″ wheel diameter. The entire housing is made of sheet steel, and covering a space of approximately 5 sq.ft., has a height of from 4 to 5 ft. Corrugated crib or plank work is the best filling to use, brush filling should be avoided. The power consumed by the circulating pump and the fan varies from 3 to 6% of the full engine power.

4. Spray Nozzles and Cooling Ponds. Spray nozzles, when the water can be supplied to them under natural head only, find application in their simplest form also in small installations. But outside of this, they are used only in the large installations because the power consumed by the nozzles when a certain higher pressure is maintained on them is considerable. Cooling ponds become available only when large unused plots of ground can be had and when the soil is of such nature that costly masonry or concrete work is not necessary. As far as the cooling of jacket water is concerned, these ponds are to-day installed only where circumstances compel their use. The waste water of scrubbers and other cleaning apparatus is, on the other hand, often made again available by the use of cooling and settling ponds.

Purity of Jacket Water is not less important than purity of feed water for boilers. Although the formation of scale occurs only in certain places of the jacket space which are highly heated, the throwing down of earthy impurities in the water forms mud or slush and other incrustations which seriously interfere with the cooling action. Water, about the quality of which there is any doubt, should never be directly supplied to the jacket, but should be first allowed to settle out all of the impurities possible in settling vessels or tanks.

To prevent the freezing of the water in the jackets when the engine is out of operation, certain chemicals, such as alcohol, etc., are sometimes added to the water. According to the experience gained in the operation of motor cars, wood alcohol is both cheap and well adapted to this purpose. The proportion used is 3 to 5%.

V. Piping

Regarding the general assembly or lay-out of pipe lines, see the assembly drawings in Part III, especially Plates VIII, IX, XXV to XXVIII, and XXX. Of course the general rules regarding the design and construction of pipe lines apply also to this case (ample cross-sections, as few bends as possible, and never sudden change of direction, means of allowing for expansion, means for draining of water and for expelling entrapped air, protection of walls of buildings against possible vibrations of any parts of the line, etc.). Care should be taken especially to see that the exhaust pipe is placed as far as possible from air and gas pipes in order to prevent the radiation from the former from heating up the fresh charge, thus reducing its density, and consequently also the engine capacity. If circumstances do not permit of a satisfactory separation of exhaust and fresh gas pipes, the effect above mentioned should be counteracted by suitable cooling arrangements, such as covering the pipes.

1. Air-pipe Line. If the volume of the fuel is small as compared with that of the air used, a condition that generally holds for oil engines, the internal diameter of the pipe may be determined from

$$d=2.2\sqrt{\frac{D^2Sn}{v}}\text{ in.}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

in which D=cylinder diameter in feet, S=stroke in feet, v=allowable air velocity in the pipe in feet per second, and n=revolutions per minute. Depending upon the length of the line, v may be assumed from 30 to 60 ft. per sec. For the latter value, which holds for lines up to about 35 ft. long, eq. (4) becomes

$$d=2.2\sqrt{\frac{D^2Sn}{60}}=.285\sqrt{D^2Sn}\text{ in.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

Lines of abnormal length should be computed by means of any of the general formulæ, taking into account pipe friction, pressure drop, etc.

If the fuel gas constitutes a considerable part of the charge, the value of d may be made correspondingly smaller (see also p. 200).

The intake end of the air-pipe line should be located in a spot cool, dry, and free from dust. The intake should be protected or screened in so that coarse dust, moisture or other undesirable additions can neither be drawn in nor reach the interior of the pipe in any other way. Regarding the muffling of the intake, see p. 296.

2. Gas-pipe Line. The internal diameter of gas pipe for the connection between engine and pressure regulator or supply valve, *for illuminating gas*, is given in Table 28, p. 304, provided the length of the pipe is not more than 25 to 35 feet.

The rest of the line from the gas bag to the main-supply pipe could be smaller, since in this pipe the gas, which is taken from the gas bag by the engine periodically, flows in a steady stream. On the other hand, this part of the line is in most cases considerably longer than the other, and on account of the added frictional resistance thus encountered, it is usual to make this pipe of somewhat larger diameter than the former.

The pressure regulator is adjusted to give a pressure of from .4 to .6″ water, measured at the gas bag. The pipe supplying the ignition flame should be branched off ahead of the gas bag, and should have $\frac{1}{4}$″ or $\frac{1}{2}$″ internal diameter.

Power-gas lines should be computed on the basis of the quantity of gas, C_h (see p. 200) that flows through the pipes at every suction stroke of the engine. For lengths not exceeding 35 to 45 feet, what has been said above regarding air-pipe lines applies equally to these producer-gas lines. For lines of considerable length the pressure drop in the pipe may be taken at from .07 to .12″ water for every 100 ft. of pipe.

For producer gas (Dowson gas) the size of the gas pipe near the engine is made at least one half of the inlet-valve cross-section. In long lines this diameter is gradually attained. Too close figuring on the sizes of gas pipe for producer-gas installations is bad, because the occasional deposits of dust and tar after a time tend to reduce the normal cross-sections of the pipe. Special care should also be taken to make the connection between engine and gas-cleaning apparatus as short as possible and with the least number of bends or changes of direction. Cleaning openings of ample size and easily accessible should be placed wherever possible, and finally, the line should be provided, near the engine, with a purge opening and a try-cock. The lines leading from the ventilating or purge opening should be of sufficient size, not under 1 inch, and should always lead directly to the open air. Even in a small installation the much-used scheme of clearing or purging the pipe line or system through the try-cock from the poor gas made during the heating-up period leads to bad contamination of the air in the room. The connecting in of cleaning apparatus in the gas-pipe line proper is not good practice, see p. 290.

3. Exhaust-pipe Line. The cross-section of the exhaust pipe from the engine to the muffler may be made from 1.1 to 1.3 times the free cross-section of the exhaust valve, depending upon the length of the connections. On the other side of the muffler the line may be somewhat smaller than the valve cross-section, provided it is straight and of ordinary length. This statement, however, does not apply to 2-cycle scavenging engines, in which the exhaust pipe should always be made as large as possible.

The discharge of exhaust gas through the chimneys of living rooms should not be permitted. In the erection of the exhaust line, attention should be paid to axial expansion, to the possibility of drawing off any water from the lowest points of the line, and to satisfactory isolation of the pipe from fuel gas and fresh-air lines as well as from inflammable material. The first division of the pipe connecting with the engine, and in installations above about 100 H.P., all of the exhaust pipe in the engine room should be water-cooled, for which purpose the water from the cylinder jackets may be used. Concerning the muffling of the exhaust, see p. 297.

4. Cooling-water Lines. Table 28 gives the diameter of the water pipe to and from the cooling tank when natural circulation is used. When a special circulating pump is used, the pipe sizes may be reduced to almost those given for fresh-water cooling in the same table. The size of the inlet pipe for fresh-water cooling, i.e., where the hot water is wasted, may be determined from

$$d=\sqrt{.023\, N_n}\text{ in.}, \qquad \dots\dots\dots\dots \qquad (6)$$

when the water consumption is placed at the liberal figure of 10 gallons per B.H.P. hour, and the velocity in the pipe is assumed at 3 ft./sec. The size of the outlet pipe is made somewhat larger, from 1.25 to 1.75 *d*, depending upon length and drop. Table 28 also gives some average figures for this case.

The branch lines leading to the other engine parts must be computed according to the quantity of water they must carry. The sum of the cross-section of all the branch pipes of course must not exceed the cross-section of the main pipe. Each branch line is furnished with its own regulating valve which is set once for all for the proper supply. When the engine is shut down, the water is shut off by a valve in the main line, while all of the valves in the branch line are left untouched. The water outlets should be free and so arranged near the engine that the hot-water temperatures can be easily read. Thorough draining of cylinder jacket and pipes is of importance on account of danger from frost and should be emphasized in the directions for operation.

TABLE 28

DIMENSIONS OF PIPE LINES FOR VARIOUS SIZES OF ENGINES

Engine Capacity, B.H.P.	Iluminating Gas Pipe.	Size of Gas Meter, Number of Lights.	Fresh Water Pipe, Inlet.	Fresh Water Pipe, Outlet.	Pipe to Cooling Tanks, Natural Circulation.	Compressed Air Pipe Line, for Starting.
	Inches.		Inches.	Inches.	Inches.	Inches.
2	¾	20	½	¾	1¼	..
4	1	30	½	¾	1½	..
6	1	40	½	⅞	1¾	..
8	1¼	50	½	⅞	1¾	..
10	1¼	60	¾	1	2	..
12	1½	80	¾	1	2¼	..
15	1½	100	¾	1	2½	..
20	2	150	¾	1	2½	1¼
25	2	150	⅞	1¼	2¾	1¼
30	2½	200	⅞	1¼	2¾	1½
35	2½	200	1	1¼	3	1½
40	3	250	1	1¼	3	1½
50	3	250	1¼	1½	3¼	1¾
60	3½	300	1¼	1½	3½	1¾
70	3½	300	1½	1¾	..	2
80	3½	350	1½	1¾	..	2
100	4	400	1¾	2	..	2¼
125	4	400	1¾	2	..	2¼
150	4½	450	2	2½	..	2½

VI. Gas Meters, Gas Bags and Pressure Regulators

These auxiliaries are used almost exclusively for illuminating-gas engines. Most gas-engine builders do not manufacture them themselves, but obtain them from special factories. For that reason a consideration of the details of construction is probably not necessary.

1. Gas Meter. Gas meters are designated according to the number of lights they will supply. Table 29 gives the approximate capacity of each size of meter in cubic feet of gas per hour. The size of the meter should be based upon the consumption at maximum engine capacity.

TABLE 29

SIZES OF GAS METERS

3 lights, capacity = approximately 17.5 cu.ft. per hr.
5 “ “ = “ 28.0 “
10 “ “ = “ 52.5 “
20 “ “ = “ 100.0 “
30 “ “ = “ 140.0 “
40 “ “ = “ 195.0 “
60 “ “ = “ 280.0 “
80 “ “ = “ 385.0 “
100 “ “ = “ 490.0 “
150 “ “ = “ 700.0 “
200 “ “ = “ 1000.0 “

The usual practice is for the gas company to furnish the meters and for the consumer to instal them.

2. Gas-pressure Regulators. The purpose of these regulators is to maintain the gas pressure in the supply pipe at a certain level, irrespective of the variation in the main pipe. They are usually set to maintain a pressure of from .5″ to .75″ water, by suitably weighting the float.

3. Gas Bags. These serve the purpose of equalizing the pressure fluctuations in the supply pipe due to the periodic aspiration of gas by the engine, and are therefore connected in directly beyond the pressure regulator, i.e., between the latter and the engine. For the smaller engines, up to say 30 H.P., a single bag gives satisfactory service; in the larger installations several of them should be connected in series, or suction reservoirs of sufficient capacity should be connected in directly beyond the pressure regulator.

The ends of the gas pipes should fill the rubber hose connections of the bag tightly, and should extend into the bag proper anywhere from 4″ to 8″. They should be strongly rounded off to prevent a cutting or wearing through of the rubber walls. The bag should not be called upon to support any part of either the inlet or outlet pipe. A good way is to use a continuous pipe passing in and out of the bag and to slot or perforate this pipe at the middle of the bag to give sufficient free passage for the gas. To protect the rubber against oil or possible external injury it is well to encase the bag in wood.

TABLE 30

APPROXIMATE DIMENSIONS OF RUBBER BAGS FOR REGULATING PRESSURE IN GAS MAINS

Capacity of Engine. B. H. P.	Width of Bag.	Length of Bag.	Length of Hose Connections.	Internal Diameter of Hose Connections.
	Inches.	Inches.	Inches.	Inches.
.5	13.0	18.0	3.25	.80
1–2	16.0	20.0	3.75	1.00
3–5	20.0	24.0	4.00	1.20
6–8	22.0	28.0	4.50	1.60
10–12	26.0	30.0	4.75	1.80
16–20	28.0	32.0	4.75	2.00
25	30.0	34.0	5.25	2.40
30	32.0	38.0	5.25	2.80

VII. General Machine Parts

The following tables give data concerning dimensions of a number of simple general machine parts which, although in constant use in the construction of gas engines, are not usually manufactured by the engine builder but bought by him in the market. The dimensions of such parts have become generally standard and the gas-engine designer therefore must, in their use, conform to general practice. For such parts as have not become generally standard, average practical values will be given.

1. Screws and Studs. Connections under heavy loads should only be made up by means of *machine screws,* because they are cheap and simple, and above all most reliable. Screws less than $\frac{5}{8}''$ in diameter may be twisted off by careless setting up, hence their use in important places should be avoided. Studs are screwed into their seats so that the length of thread in position is at least 1.5 d when placed in machinery steel and from 1.8 to $2\,d$ when screwed into cast iron or brass. The hole, when not passing all the way through, should be at least $\frac{1}{2}$ of the screw diameter deeper than this.

TABLE 31

U. S. OR SELLER'S SYSTEM OF SCREW-THREADS

BOLTS AND THREADS.						HEXAGONAL NUTS AND HEADS.					Long Diameter Square Nuts, Rough.
Diameter of Bolt.	Threads per Inch.	Diameter at Root of Thread.	Width of Flat.	Area of Bolt Body in Square Inches.	Area at Root of Thread in Square Inches.	Short Diam., Rough.	Short Diam.	Long Diam., Rough.	Thickness, Rough.	Thickness, Finish.	
Inches.		Inches.	Inches.			Inches.	Inches.	Inches.	Inches.	Inches.	Inches.
$\frac{1}{4}$	20	.185	.0062	.049	.027	$\frac{1}{2}$	$\frac{7}{16}$	$\frac{37}{64}$	$\frac{1}{4}$	$\frac{3}{16}$	$\frac{7}{10}$
$\frac{5}{16}$	18	.240	.0074	.077	.045	$\frac{19}{32}$	$\frac{17}{32}$	$\frac{11}{16}$	$\frac{5}{16}$	$\frac{1}{4}$	$\frac{10}{12}$
$\frac{3}{8}$	16	.294	.0078	.110	.068	$\frac{11}{16}$	$\frac{5}{8}$	$\frac{51}{64}$	$\frac{3}{8}$	$\frac{5}{16}$	$\frac{63}{64}$
$\frac{7}{16}$	14	.344	.0089	.150	.093	$\frac{25}{32}$	$\frac{23}{32}$	$\frac{9}{10}$	$\frac{7}{16}$	$\frac{3}{8}$	$1\frac{7}{64}$
$\frac{1}{2}$	13	.400	.0096	.196	.126	$\frac{7}{8}$	$\frac{13}{16}$	1	$\frac{1}{2}$	$\frac{7}{16}$	$1\frac{15}{64}$
$\frac{9}{16}$	12	.454	.0104	.249	.162	$\frac{31}{32}$	$\frac{29}{32}$	$1\frac{1}{8}$	$\frac{9}{16}$	$\frac{1}{2}$	$1\frac{23}{64}$
$\frac{5}{8}$	11	.507	.0113	.307	.202	$\frac{11}{16}$	1	$1\frac{7}{32}$	$\frac{5}{8}$	$\frac{9}{16}$	$1\frac{1}{2}$
$\frac{3}{4}$	10	.620	.0125	.442	.302	$1\frac{1}{4}$	$1\frac{3}{16}$	$1\frac{7}{16}$	$\frac{3}{4}$	$\frac{11}{16}$	$1\frac{49}{64}$
$\frac{7}{8}$	9	.731	.0138	.601	.420	$1\frac{7}{16}$	$1\frac{3}{8}$	$1\frac{21}{32}$	$\frac{7}{8}$	$\frac{13}{16}$	$2\frac{1}{32}$
1	8	.837	.0156	.785	.550	$1\frac{5}{8}$	$1\frac{9}{16}$	$1\frac{7}{8}$	1	$\frac{15}{16}$	$2\frac{19}{64}$
$1\frac{1}{8}$	7	.940	.0178	.994	.694	$1\frac{13}{16}$	$1\frac{3}{4}$	$2\frac{3}{32}$	$1\frac{1}{8}$	$1\frac{1}{16}$	$2\frac{9}{16}$
$1\frac{1}{4}$	7	1.065	.0178	1.227	.893	2	$1\frac{15}{16}$	$2\frac{5}{16}$	$1\frac{1}{4}$	$1\frac{3}{16}$	$2\frac{53}{64}$
$1\frac{3}{8}$	6	1.160	.0208	1.485	1.057	$2\frac{3}{16}$	$2\frac{1}{8}$	$2\frac{17}{32}$	$1\frac{3}{8}$	$1\frac{5}{16}$	$3\frac{3}{32}$
$1\frac{1}{2}$	6	1.284	.0208	1.767	1.295	$2\frac{3}{8}$	$2\frac{5}{16}$	$2\frac{3}{4}$	$1\frac{1}{2}$	$1\frac{7}{16}$	$3\frac{23}{64}$
$1\frac{5}{8}$	$5\frac{1}{2}$	1.389	.0227	2.074	1.515	$2\frac{9}{16}$	$2\frac{1}{2}$	$2\frac{31}{32}$	$1\frac{5}{8}$	$1\frac{9}{16}$	$3\frac{5}{8}$
$1\frac{3}{4}$	5	1.491	.0250	2.405	1.746	$2\frac{3}{4}$	$2\frac{11}{16}$	$3\frac{3}{16}$	$1\frac{3}{4}$	$1\frac{11}{16}$	$3\frac{57}{64}$
$1\frac{7}{8}$	5	1.616	.0250	2.761	2.051	$2\frac{15}{16}$	$2\frac{7}{8}$	$3\frac{13}{32}$	$1\frac{7}{8}$	$1\frac{13}{16}$	$4\frac{5}{32}$
2	$4\frac{1}{2}$	1.712	.0277	3.142	2.302	$3\frac{1}{8}$	$3\frac{1}{16}$	$3\frac{5}{8}$	2	$1\frac{15}{16}$	$4\frac{27}{64}$
$2\frac{1}{4}$	$4\frac{1}{2}$	1.962	.0277	3.976	3.023	$3\frac{1}{2}$	$3\frac{7}{16}$	$4\frac{1}{16}$	$2\frac{1}{4}$	$2\frac{3}{16}$	$4\frac{61}{64}$
$2\frac{1}{2}$	4	2.176	.0312	4.909	3.719	$3\frac{7}{8}$	$3\frac{13}{16}$	$4\frac{1}{2}$	$2\frac{1}{2}$	$2\frac{7}{16}$	$5\frac{31}{64}$
$2\frac{3}{4}$	4	2.426	.0312	5.940	4.620	$4\frac{1}{4}$	$4\frac{3}{16}$	$4\frac{29}{32}$	$2\frac{3}{4}$	$2\frac{11}{16}$	6
3	$3\frac{1}{2}$	2.629	.0357	7.069	5.428	$4\frac{5}{8}$	$4\frac{9}{16}$	$5\frac{3}{8}$	3	$2\frac{15}{16}$	$6\frac{17}{32}$
$3\frac{1}{4}$	$3\frac{1}{2}$	2.879	.0357	8.296	6.510	5	$4\frac{15}{16}$	$5\frac{13}{16}$	$3\frac{1}{4}$	$3\frac{3}{16}$	$7\frac{1}{16}$
$3\frac{1}{2}$	$3\frac{1}{4}$	3.100	.0384	9.621	7.548	$5\frac{3}{8}$	$5\frac{5}{16}$	$6\frac{7}{64}$	$3\frac{1}{2}$	$3\frac{7}{16}$	$7\frac{39}{64}$
$3\frac{3}{4}$	3	3.317	.0413	11.045	8.641	$5\frac{3}{4}$	$5\frac{11}{16}$	$6\frac{21}{32}$	$3\frac{3}{4}$	$3\frac{11}{16}$	$8\frac{1}{8}$
4	3	3.567	.0413	12.566	9.993	$6\frac{1}{8}$	$6\frac{1}{16}$	$7\frac{3}{32}$	4	$3\frac{15}{16}$	$8\frac{41}{64}$
$4\frac{1}{4}$	$2\frac{7}{8}$	3.798	.0435	14.186	11.329	$6\frac{1}{2}$	$6\frac{7}{16}$	$7\frac{9}{16}$	$4\frac{1}{4}$	$4\frac{3}{16}$	$9\frac{3}{16}$
$4\frac{1}{2}$	$2\frac{3}{4}$	4.028	.0454	15.904	12.743	$6\frac{7}{8}$	$6\frac{13}{16}$	$7\frac{31}{32}$	$4\frac{1}{2}$	$4\frac{7}{16}$	$9\frac{3}{4}$
$4\frac{3}{4}$	$2\frac{5}{8}$	4.256	.0476	17.721	14.226	$7\frac{1}{4}$	$7\frac{3}{16}$	$8\frac{13}{32}$	$4\frac{3}{4}$	$4\frac{11}{16}$	$10\frac{1}{4}$
5	$2\frac{1}{2}$	4.480	.0500	19.635	15.763	$7\frac{5}{8}$	$7\frac{9}{16}$	$8\frac{27}{32}$	5	$4\frac{15}{16}$	$10\frac{49}{64}$
$5\frac{1}{4}$	$2\frac{1}{2}$	4.730	.0500	21.648	17.572	8	$7\frac{15}{16}$	$9\frac{9}{32}$	$5\frac{1}{4}$	$5\frac{3}{16}$	$11\frac{23}{64}$
$5\frac{1}{2}$	$2\frac{3}{8}$	4.953	.0526	23.758	19.267	$8\frac{3}{8}$	$8\frac{5}{16}$	$9\frac{23}{32}$	$5\frac{1}{2}$	$5\frac{7}{16}$	$11\frac{7}{8}$
$5\frac{3}{4}$	$2\frac{3}{8}$	5.203	.0526	25.967	21.262	$8\frac{3}{4}$	$8\frac{11}{16}$	$10\frac{5}{32}$	$5\frac{3}{4}$	$5\frac{11}{16}$	$12\frac{3}{8}$
6	$2\frac{1}{4}$	5.423	.0555	28.274	23.098	$9\frac{1}{8}$	$9\frac{1}{16}$	$10\frac{19}{32}$	6	$5\frac{15}{16}$	$12\frac{15}{16}$

TABLE 32

ALLOWABLE LOAD IN POUNDS ON SCREWS OF VARIOUS SIZES, ACCORDING TO BACH.[1]

1	2	3	4	5	6	7	8
DIMENSIONS.		ALLOWABLE LOAD.					
External Diameter, Inches.	Area at Bottom of Thread, Square Inches	Screws under Tensile Stress only.		Screws under Tensile and Torsion Stress.		Screws set up comparatively hard in the beginning and further drawn up during operation, as Flange Bolts for example.	
		Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.
$\frac{1}{4}$	.027	185	230	138	175	32	40
$\frac{5}{16}$	.045	310	388	233	291	74	88
$\frac{3}{8}$	.068	467	583	348	435	132	165
$\frac{7}{16}$	.093	640	800	480	600	216	271
$\frac{1}{2}$	.126	825	1030	617	775	324	407
$\frac{5}{8}$	.202	1385	1730	1040	1300	623	782
$\frac{3}{4}$	.302	2070	2590	1560	1940	1070	1305
$\frac{7}{8}$	.420	2880	3600	2150	2700	1610	2020
1	.550	3780	4730	2830	3530	2110	2650
$1\frac{1}{8}$	.694	4750	5950	3570	4450	2680	3320
$1\frac{1}{4}$	.893	6100	7640	4580	5730	3430	4280
$1\frac{3}{8}$	1.057	7200	9060	5420	6790	4070	5090
$1\frac{1}{2}$	1.295	8870	11050	6650	8330	4980	6230
$1\frac{5}{8}$	1.515	10000	12550	7550	9400	5650	7060
$1\frac{3}{4}$	1.746	11900	14900	8950	11700	6740	8400
$1\frac{7}{8}$	2.051	13550	16920	10100	13300	7620	9550
2	2.302	15750	19650	11800	15450	8850	11100
$2\frac{1}{4}$	3.023	20000	25000	14950	19550	11200	14050
$2\frac{1}{2}$	3.719	25400	31900	19000	24900	14300	17900
$2\frac{3}{4}$	4.620	30400	38100	22800	29800	17100	21400
3	5.428	37100	46500	27800	36500	20900	26000

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[1] The original table was based on Whitworth thread, but the figures have been directly transposed, since the differences between the bottom areas of this and the Seller's thread are not great.

NOTE. Columns 3, 5, and 7 apply to screws of steel of average quality; columns 4, 6, and 8 to screws of superior grade. The stress per square inch of area at bottom of thread has been taken as follows:

Column.......	3	4	5	6	7	8
	6800	8500	5100	6400	3820	4650 lbs. per sq.in.,

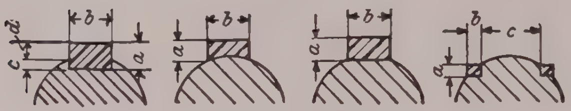
except for screws from $\frac{1}{4}$ in. to $\frac{3}{4}$ in., for which the stress has in each case been taken 10% less. Where first-class soft steel is employed the values in columns 7 and 8 may be exceeded 30%.

For gas thread, see Table 34, p. 309.

2. Keys. The material used is soft or crucible steel. Keys are tapered along the top surface only and the taper amounts to from $\frac{1}{75}$ to $\frac{1}{100}$. When two sunk, flat, or saddle keys are employed they are placed at an angle of 120°, unless peculiarities of hub construction, etc., compel the use of smaller angles.

Table 33

DIMENSIONS OF KEYS



Figs. 467*a*–467*d*.

Diameter of Shaft.	Flat or Saddle Keys.		Sunk Keys.				
	Width, *b*.	Thickness, *a*.	Keys.		Key Seat, Depth in		
			Width, *b*.	Thickness, *a*.	Shaft.	Bore.	
Inches.	Inches.	Inches.	Inches.	Inches.	Inches.	Inches.	Distance between double keys $c \sim \frac{1}{2}$ shaft diameter. For other examples of such construction see Fig. 276, p. 209, Fig. 303, p. 233, and Fig. 312, p. 237.
.75	.40	.20	.40	.25	.10	.15	
1.00	.45	.20	.45	.25	.10	.15	
1.25	.50	.20	.50	.30	.10	.20	
1.50	.55	.25	.55	.35	.10	.25	
1.75	.65	.25	65	.40	.125	.275	
2.00	.70	.25	.70	.40	.125	.275	
2.25	.80	.25	.80	.45	.15	.30	
2.50	.85	.30	.85	.50	.15	.35	
2.75	.90	.30	.90	.50	.15	.35	
3.00	.95	.35	.95	.55	.15	.40	
3.50	1.10	.35	1.10	.60	.20	.40	
4.00	1.20	.50	1.20	.70	.20	.50	
4.50	Suitable for light shafts only.		1.35	.75	.20	.55	
5.00			1.50	.80	.25	.55	
6.00	Double Keys.		1.70	.95	.30	.65	
7.00			2.00	1.10	.35	.75	
8.00	1.00	.80	2.25	1.20	.40	.80	
9.00	1.15	.90	2.50	1.45	.45	1.00	
10.00	1.30	1.00	2.75	1.65	.50	1.15	
12.00	1.55	1.25	3.20	2.00	.55	1.45	
14.00	1.75	1.40	3.70	2.25	.70	1.55	
16.00	1.95	1.50	4.10	2.45	.80	1.65	
18.00	2.10	1.60	4.50	2.60	.90	1.70	
20.00	2.20	1.70	4.80	2.80	1.00	1.80	

3. Gas Pipe, Pipe Fittings, and Flanges.

TABLE 34—DIMENSIONS OF WROUGHT-IRON WELDED PIPE—U. S. Standard

Nominal Inside Diameter.	Actual Outside Diameter.	Actual Inside Diameter.	Thickness of Metal.	Internal Area.	External Area.	Weight of Pipe per Lineal Foot.	Number of Threads per Inch.
Inches.	Inches.	Inches.	Inches.	Square Inches.	Square Inches.	Lbs.	No.
$\frac{1}{8}$	.405	.270	.068	.057	.1288	.24	27
$\frac{1}{4}$	.540	.364	.085	.104	.2290	.42	18
$\frac{3}{8}$	.675	.493	.091	.191	.3578	.56	18
$\frac{1}{2}$	.840	.622	.109	.304	.554	.84	14
$\frac{3}{4}$	1.050	824	.113	.533	.866	1.12	14
1	1.315	1.048	.134	.861	1.358	1.67	$11\frac{1}{2}$
$1\frac{1}{4}$	1.660	1.380	.140	1.496	2.164	2.24	$11\frac{1}{2}$
$1\frac{1}{2}$	1.900	1.610	.145	2.036	2.835	2.68	$11\frac{1}{2}$
2	2.375	2.067	.154	3.356	4.430	3.61	$11\frac{1}{2}$
$2\frac{1}{2}$	2.875	2.468	.204	4.780	6.492	5.74	8
3	3.500	3.067	.217	7.383	9.621	7.54	8
$3\frac{1}{2}$	4.000	3.548	.226	9.887	12.566	9.00	8
4	4.500	4.026	.237	12.730	15.904	10.66	8
$4\frac{1}{2}$	5.000	4.508	.246	15.961	19.635	12.34	8
5	5.563	5.048	.259	19.986	24.301	14.50	8
6	6.625	6.065	.280	28.890	34.472	18.76	8
7	7.625	7.023	.301	38.738	45.664	23.27	8
8	8.625	7.981	.322	50.027	58.426	28.18	8
9	9.625	8.937	.344	62.730	72.760	33.70	8
10	10.75	10.018	.366	78.823	90.763	40.06	8
11	11.75	11.000	.375	95.033	108.434	45.02	8
12	12.75	12.000	.375	113.098	127.677	49.00	8
13	14	13.25	.375	137.887	153.938	54.00	8
14	15	14.25	.375	159.485	176.715	58.00	8
15	16	15.25	.375	182.665	201.062	62.00	8
..	18	17.25	.375	239.706	254.470	70.00	..
..	20	19.25	.375	291.040	314.159	78.00	..
..	22	21.25	.375	354.657	380.134	85.00	..
..	24	23.25	.375	424.558	452.390	93.00	..

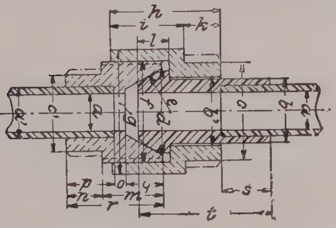


FIG. 468.

TABLE 35

DIMENSIONS OF BRASS UNIONS FOR HIGH-PRESSURE COPPER PIPE

ALL DIMENSIONS IN INCHES

Material of the union itself is brass or bronze, that of the cone is copper

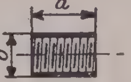
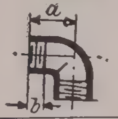
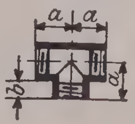
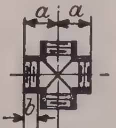
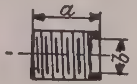
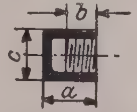
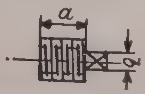
Diameter of Pipe. Internal.	Diameter of Pipe. External.			Small Diameter of Hexagon.																	
a	*a'*	*b*	*b'*	*c*	*c'*	*d*	*e*	*f* *	*g*	*h*	*i*	*k*	*l*	*m*	*n*	*o*	*p*	*q*	*r*	*s*	*t*
.16	.28	.44	.48	.92	.76	.68	.32	$\frac{3}{4}''$	1.44	1.80	1.20	.60	.28	1.00	.60	With seat angle =60°, the cone continues straight to the bore of the tube up to $a=.56$			1.60	.60	1.72
.28	.44	.60	.64	1.08	.92	.84	.40	$\frac{7}{8}''$	1.60	1.80	1.20	.60	.36	1.00	.60				1.60	.60	1.80
.40	.56	.72	.76	1.32	1.08	1.00	.48	1″	1.76	1.80	1.20	.60	.40	1.00	.60				1.60	.60	1.84
.48	.64	.84	.88	1.32	1.32	1.16	.56	$1\frac{1}{8}''$	1.92	1.80	1.20	.60	.48	1.00	.60				1.60	.80	2.12
.56	.76	.96	1.00	1.44	1.32	1.32	.64	$1\frac{1}{4}''$	2.08	1.80	1.20	.60	.52	1.00	.60				1.60	.80	2.16
.64	.84	1.04	1.08	1.60	1.44	1.40	.68	$1\frac{1}{4}''$	2.08	1.80	1.20	.60	.52	1.00	.60	.20	.80	.60	1.60	.80	2.16
.80	1.00	1.20	1.24	1.80	1.60	1.60	.76	$1\frac{1}{2}''$	2.32	1.80	1.20	.60	.52	1.00	.80	.20	1.00	.60	1.80	1.00	2.16
1.00	1.20	1.44	1.48	2.00	1.80	1.88	.88	$1\frac{3}{4}''$	2.64	1.88	1.20	.60	.56	1.00	1.00	.20	1.20	.60	2.00	1.20	2.20
1.20	1.40	1.68	1.72	2.16	2.00	2.16	1.00	2	2.88	1.88	1.20	.60	.60	1.00	1.20	.20	1.40	.60	2.20	1.40	2.24

* Pipe Thread

TABLE 36

DIMENSIONS AND WEIGHT OF STANDARD PIPE FITTINGS

Compiled mostly from data given by CRANE Co.

Nominal size.	. . . . . . inches	$\frac{1}{8}$	$\frac{1}{4}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{3}{4}$	1	$1\frac{1}{4}$	$1\frac{1}{2}$	2	$2\frac{1}{2}$	3	$3\frac{1}{2}$	4
Couplings 	Length. . . . . *a*	$\frac{13}{16}$	1	$1\frac{3}{32}$	$1\frac{5}{16}$	$1\frac{9}{16}$	$1\frac{3}{4}$	$1\frac{15}{16}$	$2\frac{3}{16}$	$2\frac{5}{16}$	$2\frac{11}{16}$	3	$3\frac{1}{8}$	$3\frac{1}{8}$
	Outside diameter . . . . . *b*	$\frac{19}{32}$	$\frac{25}{32}$	$\frac{29}{32}$	$1\frac{3}{32}$	$1\frac{11}{32}$	$1\frac{21}{32}$	2	$2\frac{9}{32}$	$2\frac{13}{16}$	$3\frac{9}{32}$	$4\frac{1}{64}$	$4\frac{1}{2}$	$5\frac{3}{16}$
	Weight of 10. . . . . . lbs.	.31	.62	.93	1.56	2.50	3.75	5.60	8.15	11.25	16.20	30.60	32.50	42.50
Elbows 	Length of branch. . . . . *a*	$\frac{5}{8}$	$\frac{13}{16}$	$\frac{15}{16}$	$1\frac{1}{8}$	$1\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{3}{4}$	$1\frac{7}{8}$	$2\frac{1}{4}$	$2\frac{3}{4}$	$3\frac{3}{16}$	$3\frac{7}{16}$	$3\frac{3}{4}$
	Length of thread. . . . . *b*	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{21}{32}$	$\frac{23}{32}$	$\frac{25}{32}$	$\frac{7}{8}$	$\frac{15}{16}$	$1\frac{1}{32}$	$1\frac{1}{8}$	$1\frac{3}{16}$
	Weight of 10. . . . . . lbs.	.70	1.50	2.20	3.80	6.50	9.50	15.00	20.00	32.00	56.00	75.00	94.00	125.00
T's 	Length of branch. . . . . *a*	$\frac{5}{8}$	$\frac{13}{16}$	$\frac{15}{16}$	$1\frac{1}{8}$	$1\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{3}{4}$	$1\frac{7}{8}$	$2\frac{1}{4}$	$2\frac{3}{4}$	$3\frac{3}{16}$	$3\frac{7}{16}$	$3\frac{3}{4}$
	Length of thread. . . . . *b*	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{11}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{21}{32}$	$\frac{23}{32}$	$\frac{25}{32}$	$\frac{7}{8}$	$\frac{15}{16}$	$1\frac{1}{32}$	$1\frac{1}{8}$	$1\frac{3}{16}$
	Weight of 10. . . . . . lbs.	.80	2.00	2.50	5.00	10.00	14.00	21.00	27.00	46.00	78.00	105.00	130.00	180.00
Crosses 	Length of branch. . . . . *a*	$\frac{5}{8}$	$\frac{13}{16}$	$\frac{15}{16}$	$1\frac{1}{8}$	$1\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{3}{4}$	$1\frac{7}{8}$	$2\frac{1}{4}$	$2\frac{3}{4}$	$3\frac{3}{16}$	$3\frac{7}{16}$	$3\frac{3}{4}$
	Length of thread. . . . . *b*	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{21}{32}$	$\frac{23}{32}$	$\frac{25}{32}$	$\frac{7}{8}$	$\frac{15}{16}$	$1\frac{1}{32}$	$1\frac{1}{8}$	$1\frac{13}{16}$
	Weight of 10. . . . . . lbs.	1.10 appr.	2.50	3.00	5.00	11.00	17.00	25.00	34.00	56.00	95.00	125.00	160.00	210.00
Close nipples 	Length. . . . . *a*	$\frac{3}{4}$	$\frac{7}{8}$	1	$1\frac{1}{8}$	$1\frac{3}{8}$	$1\frac{1}{2}$	$1\frac{5}{8}$	$1\frac{3}{4}$	2	$2\frac{1}{2}$	$2\frac{1}{2}$	$2\frac{3}{4}$	3
	Internal diameter. . . . . *b*	.270	.364	.494	.623	.824	1.048	1.380	1.611	2.067	2.468	3.067	3.548	4.026
Caps  Diam. and weights appr. only	Length. . . . . *a*	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{13}{16}$	1	1	$1\frac{1}{16}$	$1\frac{5}{16}$	$1\frac{11}{16}$	$1\frac{3}{4}$	$1\frac{3}{4}$	$1\frac{13}{16}$	$2\frac{1}{4}$
	Length of thread. . . . . *b*	$\frac{1}{4}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{13}{16}$	$\frac{12}{16}$	1	1	1	$1\frac{1}{16}$	$1\frac{1}{8}$
	External diameter. . . . . *c*	$\frac{9}{16}$	$\frac{25}{32}$	$\frac{15}{16}$	$1\frac{1}{16}$	$1\frac{5}{16}$	$1\frac{21}{32}$	2	$2\frac{3}{16}$	$3\frac{1}{8}$	$3\frac{5}{16}$	$4\frac{1}{2}$	$5\frac{1}{8}$	$5\frac{1}{2}$
	Weight of 10. . . . . . lbs.	.27	.52	.76	1.12	1.92	3.14	5.80	7.20	11.00	19.20	38.30	55.0	68.8
Plugs  Diam. and weights appr. only	Length. . . . . *a*	$\frac{7}{16}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{19}{32}$	$\frac{5}{8}$	$\frac{11}{16}$	$\frac{3}{4}$	$\frac{25}{32}$	$\frac{13}{16}$	$1\frac{1}{3}$	$1\frac{1}{8}$	$1\frac{3}{16}$	$1\frac{1}{4}$
	Side of square head. . . . *b*	$\frac{9}{32}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{15}{16}$	1	$1\frac{1}{4}$	$1\frac{11}{16}$	$1\frac{7}{8}$	$2\frac{1}{16}$
	Weight of 10. . . . . . lbs.	.16	.30	.50	.82	1.18	1.76	3.36	1.27	6.66	12.10	17.00	25.00	34.20

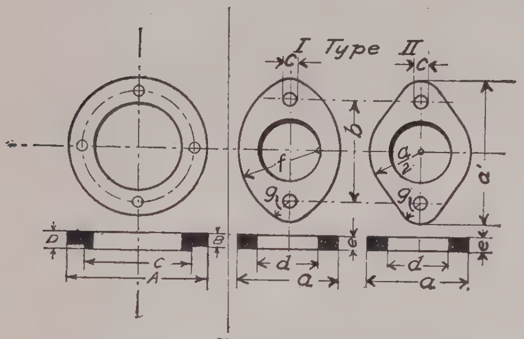


FIG. 469.

TABLE 37

DIMENSIONS OF STANDARD PIPE FLANGES

(Millar & Son)

ALL DIMENSIONS IN INCHES

Size of Pipe, Inches.	External Diameter. *A*	Thickness. *B*	Diameter of Hub. *C*	Length of Thread. *D*	Diameter of Bolt Circle.	Number of Bolts.	Size of Bolts.	Length of Bolts for Standard.
1	4	$\frac{7}{16}$	$1\frac{15}{16}$	$\frac{11}{16}$	3	4	$\frac{7}{16}$	$1\frac{1}{2}$
$1\frac{1}{4}$	$4\frac{1}{2}$	$\frac{1}{2}$	$2\frac{5}{16}$	$\frac{3}{4}$	$3\frac{3}{8}$	4	$\frac{7}{16}$	$1\frac{1}{2}$
$1\frac{1}{2}$	5	$\frac{9}{16}$	$2\frac{5}{8}$	$\frac{7}{8}$	$3\frac{7}{8}$	4	$\frac{1}{2}$	$1\frac{3}{4}$
2	6	$\frac{5}{8}$	$3\frac{1}{8}$	1	$4\frac{3}{4}$	4	$\frac{5}{8}$	2
$2\frac{1}{2}$	7	$\frac{11}{16}$	$3\frac{5}{8}$	$1\frac{1}{16}$	$5\frac{1}{2}$	4	$\frac{5}{8}$	$2\frac{1}{4}$
3	$7\frac{1}{2}$	$\frac{3}{4}$	$4\frac{5}{16}$	$1\frac{1}{8}$	6	4	$\frac{5}{8}$	$2\frac{1}{2}$
$3\frac{1}{2}$	$8\frac{1}{2}$	$\frac{13}{16}$	$4\frac{7}{8}$	$1\frac{3}{16}$	7	4	$\frac{5}{8}$	$2\frac{1}{2}$
4	9	$\frac{15}{16}$	$5\frac{3}{8}$	$1\frac{3}{16}$	$7\frac{1}{2}$	4	$\frac{3}{4}$	$2\frac{3}{4}$
$4\frac{1}{2}$	$9\frac{1}{4}$	$\frac{15}{16}$	$5\frac{13}{16}$	$1\frac{1}{4}$	$7\frac{3}{4}$	8	$\frac{3}{4}$	3
5	10	$\frac{15}{16}$	$6\frac{7}{16}$	$1\frac{5}{16}$	$8\frac{1}{2}$	8	$\frac{3}{4}$	3
6	11	1	$7\frac{9}{16}$	$1\frac{7}{16}$	$9\frac{1}{2}$	8	$\frac{3}{4}$	3
7	$12\frac{1}{2}$	$1\frac{1}{16}$	$8\frac{5}{8}$	$1\frac{1}{2}$	$10\frac{3}{4}$	8	$\frac{3}{4}$	$3\frac{1}{4}$
8	$13\frac{1}{2}$	$1\frac{1}{8}$	$9\frac{11}{16}$	$1\frac{5}{8}$	$11\frac{3}{4}$	8	$\frac{3}{4}$	$3\frac{1}{2}$
9	15	$1\frac{1}{8}$	$10\frac{5}{8}$	$1\frac{3}{4}$	$13\frac{1}{4}$	12	$\frac{3}{4}$	$3\frac{1}{2}$
10	16	$1\frac{3}{16}$	$11\frac{15}{16}$	$1\frac{7}{8}$	$14\frac{1}{4}$	12	$\frac{7}{8}$	$3\frac{3}{4}$

TABLE 38

DIMENSIONS SUITABLE FOR OVAL FLANGES

ALL DIMENSIONS IN INCHES

Size of pipe	$\frac{1}{4}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	1	$1\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{3}{4}$	2	$2\frac{1}{2}$	3
Width of flange, *a*	1.40	1.60	1.92	2.08	2.32	2.64	3.04	3.36	3.76	4.10	4.80	5.56
Length of flange, *a′*	2.64	2.80	3.44	3.60	3.76	4.00	4.64	4.96	5.46	6.16	6.90	7.76
Center to center of holes, *b*	1.60	1.68	2.16	2.32	2.48	2.72	3.20	3.44	3.76	4.24	4.87	5.60
Diameter of bolt hole, *c*	$\frac{7}{16}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{5}{8}$	$\frac{5}{8}$	$\frac{13}{16}$	$\frac{13}{16}$	$\frac{15}{16}$
Thickness of flange, *e**	.36	.36	.36	.40	.40	.40	.40	.40	.50	.50	.60	.72
Radius, *f* †	2.32	2.16	2.68	2.52	2.42	2.36	2.72	2.80	3.08	3.60	3.84	4.24
Radius, *g*	.52	.56	.64	.64	.64	.64	.72	.76	.84	.96	1.00	1.08

* Applies to steel flanges, for cast iron flanges, thickness = appr. $1\frac{2}{3}e$.

† Applies to Type I, for Type II mean radius $= \frac{a}{2}$.

4. Helical Springs. In the following table, which was printed in *Power*, July 1907, P represents the maximum safe load for helical springs made of round-steel wire, D is the pitch diameter of the spring in inches to the center of the wire, and f the corresponding deflection of the spring (tension or compression) in inches for one coil.

The table is based upon the following general equations:

$$P=\frac{\pi}{8}\frac{\delta^3}{D}K_d=.392\frac{\delta^3}{D}K_d \text{ lbs.},\qquad f=\frac{64r^3P}{\delta^4E}i=\frac{8\,D^3P}{\delta^4E}i \text{ in.}$$

δ = diameter of the wire in inches;
E = torsional modulus, assumed at 12600000;
i = number of windings = 1.0 in the table;
K_d = safe fiber stress, has been assumed to vary for the different sizes of wire as shown in the table.[1]

Regarding the determination of the principal dimensions of valve springs, see p. 203; concerning the law for the variation of load with deflection, see the force diagram, Fig. 257.

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[1] The computer of this table states that it has been adopted by a number of designers and found useful. It seems to the translator, however, that the maximum fiber stress assumed is rather high for all sizes of wire from ¼″ to 1½″. In his own practice he would use about 60 000 lbs. for the 1½″ wire, and increase it by steps up to 100 000 lbs. for the ¼″ wire. If it is decided to use any other stress than that assumed in the table, the corrections for P and f are very quickly made, for since K_d occurs in the first power in the equation for P, and P occurs in the first power in the equation for f, it is only necessary to multiply the figures in the table by the ratio of the stresses, taken in the proper sense. For instance

From the table, for a stress assumed at 90 000 lbs., $\delta=1.0$, and pitch diameter $=6.0''$, $P=5870$, $f=.805''$.

If the stress had been assumed at 60 000 lbs., then

$$P=\frac{60\,000}{90\,000}5870=3900 \text{ lbs., and } f=\frac{60\,000}{90\,000}.805=.536''.$$

PART III

CONSTRUCTION, ERECTION AND TESTS OF MODERN INTERNAL-COMBUSTION ENGINES

The following pages contain some detailed information on a selected number of the most recent types of internal-combustion engines, including in most cases results of the latest tests, together with lists of sizes and weights. Although the constructions discussed cannot perhaps in all cases be regarded as typical without qualification, taking them altogether they represent a very gratifying picture of the high development the gas-engine industry has to-day attained.

The drawings, clear in execution and true to detail, hardly call for any extended descriptions. It is sufficient to point out the main details of the design, from which the manner of operation will in general be easily understood. In view of the fact that many of these types are used for oil as well as for gas fuel, the engines described in the following pages are not classified in any way according to fuel used, but the subject-matter is arranged simply according to makers. The sequence has been decided, as far as possible, only with reference to the length of time the various engines have been on the market, there being absolutely no intention to establish an order of merit.[1]

A. STATIONARY ENGINES

The great majority of stationary internal-combustion engines serve simply as prime movers operating single machines or groups of machines by means of main shafting, countershafting, and other types of transmission. Direct connection of power consumer to prime mover is a special construction of which direct-connected electric generators, blowing and pumping engines are well-known examples.

I. Capital Cost and Cost of Erection

The question of *operating costs* touches the designer only indirectly. These costs are made up not only of fuel costs, cost of lubrication and attendance, but are strongly influenced also by cost of erection, of maintenance, floor space required, etc. For that reason a consideration of the latter factors is therefore not out of place. General comparisons of operating costs made up on the usual plan are nearly worthless. The items fundamentally concerned in computations of this kind differ so much among themselves, that only a close examination of every item in each separate case will serve the purpose. It is hoped that the following pages may serve as a reliable guide to this end.

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[1] Translator's Note. This does not apply to the material added describing American machines. This was written up simply in the order in which the information was received.

1. Fuel Costs. The statement of fuel consumption should include that used during starting up, during stand-by periods, etc. The purchase price of the fuel is understood to refer to the actual cost at the place of consumption, that is, freight, loss through drying and sifting, and, in the case of gas, the influence of pressure and temperature upon meter readings should also be taken into account. Where the load on the plant changes considerably, it is usually found that only the total fuel used per operating day is known, and not the quantity per H.P. hour. In general, the fuel consumption per horse-power changes with the load on the engine, that is, increases or decreases with the latter in an inverse ratio. Different engines vary considerably in this respect, but the figures in the following table serve as practical average figures for the increase of fuel consumption with decrease of load:

Load approximately	$\frac{3}{4}$	$\frac{2}{3}$	$\frac{1}{2}$	$\frac{1}{3}$	$\frac{1}{4}$ of rated load	
Illuminating-gas engine, about	10	20	35	60	90%	greater consumption per B.H.P. hour than at full load.
Suction gas engine, about	20	30	50	75	100%	
Diesel oil engine, about	10	20	30	55	80%	

Gas engines in combination with gas producers show a more rapid increase in the fuel consumption with decrease in load than either illuminating gas or oil engines. For this reason the latter may at certain loads be more efficient than the former when at full load the reverse is the case.

If an engine uses on the average q B.T.U. per B.H.P. hour, and if 1000 B.T.U. cost b cents at the place of consumption, the fuel costs per B.H.P. will be

$$K_b = \frac{q}{1000} b \text{ cents.} \quad \ldots \ldots \ldots \quad (1)$$

If the economic efficiency of the installation is $\eta_w = \frac{2545}{q}$, so that $q = \frac{2545}{\eta_w}$, we may also write

$$K_b = \frac{2545}{\eta_w} \cdot \frac{b}{1000} = \frac{2.545 b}{\eta_w} \text{ cents.} \quad \ldots \ldots \ldots \quad (2)$$

Examples. 1. Suppose that a total efficiency of 18% is guaranteed for a 50-H.P. suction-gas installation. Anthracite delivered at the producer house costs $6.80 per ton. The heating value of the fuel is 13 500 B.T.U. per pound. We then have

$$q = \frac{2545}{.18} = 14\,120 \text{ B.T.U.,} \quad b = \frac{680 \times 1000}{2000 \times 13\,500} = .025 \text{ cent,}$$

and

$$K_b = \frac{2545}{.18} \times \frac{.025}{1000} = \frac{2.545 \times .025}{.18} = .35 \text{ cent.}$$

2. Cost of Attendance. The statement often made that a given gas-engine installation needs no attendance is of course untrue, for even the smallest size requires some attention. It is however certain that gas-engine plants throughout require less attendance than steam plants of the same capacity. They require less time, or may be served by cheaper labor. The amount of attendance required of course directly depends upon the brake horse-power N_n of the installation. For the ordinary operating period of ten hours per day the hours of labor that must be expended on the installation may be approximately expressed by

$$W = .25\sqrt{N_n} \text{ hours (for illuminating gas and good grade oil engines)} \quad . \quad . \quad (3)$$

and by $$W = 1.25\sqrt{N_n} \text{ hours (for anthracite or coke-suction gas plants).} \quad . \quad . \quad . \quad . \quad . \quad (4)$$

If the cost of labor, wages of the attendant, is x cents per hour, the cost of labor per B.H.P. hour, based on the normal rating of the engine, will then be

$$K_w = \frac{.25\sqrt{N_n}}{10N_n}x = \frac{.025x}{\sqrt{N_n}} \text{ cents (for illuminating gas and good grade oil engines)} \quad . \quad (3a)$$

and

$$K_w = \frac{1.25\sqrt{N_n}}{10N_n}x = \frac{.125x}{\sqrt{N_n}} \text{ cents (for anthracite and coke suction-gas plants)} \quad . \quad . \quad . \quad (4a)$$

Assuming an average wage scale of 20 cents per hour, the cost of attendance per B.H.P. hour would figure out

$$\text{For illuminating-gas engines, } K_w = \frac{.50}{\sqrt{N_n}} \text{ cents} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

$$\text{For suction-gas engines, } K_w = \frac{2.5}{\sqrt{N_n}} \text{ cents.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5a)$$

Example. 2. The 50 H.P. plant mentioned in the first example requires a daily attendance equal to $W = 1.25\sqrt{50} \sim 9$ hours, which, with the above assumption, makes the cost of labor per B.H.P. hour equal to $K_w = \frac{2.5}{\sqrt{50}} = .35$ cent.

As soon as W exceeds ten or twelve hours, the total time should be distributed between two or more men. Taking a 100 H.P. suction-gas plant, for instance, for which $W = 1.25\sqrt{100} = 12.5$ hours, one man could probably do the work; a 500 H.P. plant, on the other hand, should have three men, since $W = 1.25\sqrt{500} = 28$. In large installations, consisting, say, of n engine units with their producers, the required time of attendance may be approximately found from

$$\left.\begin{array}{l} W = 1.25\sqrt{nN_n} \text{ hours} \\ \text{or preferably} \quad W = 1.25n\sqrt{nN_n} \text{ hours} \end{array}\right\} \text{per operating day of the entire installation.}$$

If, in cases where W is considerably less than 10–12 hours, the remaining time of the attendant cannot be used in other work, his entire wages must of course be charged against the plant.

3. Cost of Lubrication and of Cleaning Material. This item can only be estimated on the basis of previous experience. As in every other power plant, the cost of oil and of waste is considerably reduced when the oil is recovered and filtered and the waste is washed for continued use. In this way the expenditure for these two materials may be brought very close to the figures obtaining in steam plants of the same size. In medium-sized engines the consumption of lubricating oil is approximately from .005 to .008 pint per B.H.P. hour; in favorable cases, however, it may be only .004 pint and less.

4. Interest (3 to 4.5%) **and Depreciation** (7 to 10%), **Computed on the First Cost** (A_m) **of the Engine** *together with cost of auxiliaries, and including freight charges, cost of foundation and cost of erection, but excluding the cost of the building.* The cost of the ground occupied is usually left out of consideration on the assumption that its value increases with time. It should, however, be remarked that under certain circumstances the cost of acquiring the ground may constitute a large part of the cost of the installation. Consequently the size of building space required by a given engine installation is of some importance as far as operating costs are concerned, although in comparative estimates usually only the costs of buildings and foundations appear. The latter of course increase with the floor space covered.[1]

5. Interest (3 to 4½%) **and Depreciation** (2 to 3%), **Computed on the First Cost** (A_g) **of the Building.** *Where the building or the room is not owned, the rent is substituted for this item.* If in the same building other machines besides the engine installation are also in operation, in computing the operating costs the engine plant is charged up with a proportionate part of the sum A_g.

Placing the interest charge at c%, and the depreciation at a%, and assuming that the average power developed by the installation is N_e, the amount chargeable against the plant per horse-power per year on this basis will be

$$K=\frac{A}{100N_e}(c+a) \text{ dollars}, \quad \ldots\ldots\ldots\ldots \quad (6)$$

---

[1] The following figures give a case in point:

(*a*) Horizontal suction-gas engine, 100 H.P., with producer, wet and dry purifiers, etc., two rooms covering together 1400 sq.ft.

(*b*) Vertical suction-gas engine, 100 H.P., with producer, wet and dry purifiers, tar extractor, etc., two rooms covering together 970 sq.ft.

Difference in the floor space is 430 sq.ft. (approx. 30%) which decreased the cost of the space approximately $3750. The building for installation (*a*) costs approximately 1400×$1.40=$1960; for installation (*b*) approximately 970×$1.40=$1360. Difference in the cost of buildings, excluding foundation, is $600. The vertical engine installation is therefore in all about $4350 cheaper than the horizontal, all other things being the same.

The cost of all the machinery, erected and ready for operation, is in both cases $8000. Hence, according to the above figures, fully one half of this price may be paid out of the saving in cost of ground and buildings resulting from the adoption of plan (*b*). The cost of floor space was $8.70 per sq.ft., which is not unusually high. In large cities this often reaches $10 and $12 per sq.ft., and although commercial plants are not often installed in the select quarters warranting this price, it not infrequently happens that in the proposed erection of plants supplying city blocks such prices are encountered.

[TRANSLATOR'S NOTE. The above figures apply of course primarily to German conditions.]

where A represents first cost, which may be either A_m = first cost of engine and auxiliaries, as above noted, or A_g = first cost of building.

If Z is the number of hours the plant is in operation per year, then

$$K_m=\frac{A_m}{100N_e}\frac{(c+a)}{Z}\text{ dollars per B.H.P. hour} \quad . \; . \; . \; . \; . \; . \; . \; . \quad (7)$$

and

$$K_g=\frac{A_g}{100N_e}\frac{(c+a)}{Z}\text{ dollars per B.H.P. hour.} \quad . \; . \; . \; . \; . \; . \; . \quad (7a)$$

Assuming as usual that

$$c=4\% \quad \text{and} \quad a=8\% \text{ for } A_m$$

and

$$c=4\% \quad \text{and} \quad a=2\% \text{ for } A_g$$

and taking $Z=3000$ hours per annum, eqs. (7) and (7*a*) reduce to

$$K_m=\frac{A_m}{250N_e}\text{ cents per B.H.P. hour} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad (8)$$

and

$$K_g=\frac{A_g}{500N_e}\text{ cents per B.H.P. hour.} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad (8a)$$

From these equations it appears that the greater the product of N_e and Z, the lower the cost per H.P. hour, which shows that both high load factors and long operating hours are desirable. The same thing applies with even greater force to the fuel costs, since these (see under **1**) increase much more rapidly with a decrease in N_e than either K_m or K_g. As between the individual effects of the factors N_e and Z, it would seem better to operate a plant for a short time at high load factor than for longer hours under a lower load. It is also better to shut down a plant for a day at a time than for an hour now and then, on account of smaller stand-by losses.

Finally, an inspection of eqs. (7) and (7*a*) will also show that the bad effect of short operating hours upon K_m and K_g can in a measure be counteracted by small outlay in first cost (A_m or A_g). In other words, all other conditions being the same, the less a given plant is used the lower should be its first cost. It is possible of course to reach the same end by proportionately decreasing the charge (a) for depreciation, but that can only be considered when the interruptions in the operation occur at regular intervals, when the sum total of the stand-by periods constitutes a large part of the year, and when during this time the plant is maintained in its best condition.

6. Maintenance and Repair of Engines and Auxiliaries (2 to 3% of the first cost A_m). The quality of the construction and especially the degree of "hardiness" against the normal occurrences of ordinary operation play an important part in the determination of this item, a fact which should not be lost sight of, especially when judging suction-gas plants. In the latter case, poor scrubbing arrangements, causing a rapid scoring of the cylinder, bad vaporizer construction, or rapid wear on producer lining and grate, may, under certain circumstances, cause a repair bill which exceeds even 10 or 15% of the first cost. Under normal conditions, however, a charge of 2 or 3% is quite sufficient. Extraordinary occurrences, like accidents due to careless attendance, etc., are of course not considered in the figure.

7. Maintenance and Repair on Buildings (1 to 1½% of the first cost A_g). Strictly speaking, another item that should be taken into account is insurance. But the latter charge is very nominal when reduced to the horse-power basis, and is consequently usually neglected.

In making estimates of operating costs, items **2** to **7** are very often considered as merely incidental expenses, making the cost of fuel the only criterion of economy. This method, however, applies only in cases where the cost of fuel is very high (illuminating gas, and, in part, also crude oil distillates); where the fuel is cheaper or is utilized at very high efficiency it will nearly always be found that the "incidental" expenses constitute the greater part of the total operating costs. Fuel costs alone therefore never give a certain indication regarding the economic status of a given plant. Plants high in first cost, or those in which the repair and maintenance account is high, or those which require constant close attention, occupy much floor space or call for expensive foundations, all of these may under certain conditions operate with small economy, although the efficiency at which the fuel is burned may be of the highest.

Table 40, and the diagram, Fig. 470, derived from it, gives a convenient survey of all of the items concerned in the computation of operating costs. The figures are based upon unit horse-power, because it is easier to compare various sizes and types of engines better on this basis than if the yearly cost had been made the unit. The table has been constructed by the use of the equations given above, the assumptions for A_m and A_g, as well as for fuel costs and incidental expenses, being based upon the best obtainable average figures for modern *horizontal* installations. In order to simplify matters, the average capacity N_e developed by the plant is put equal to the normal rating N_n, and the time of operation has been assumed at 3000 hours per year. Where these assumptions do not apply, it is easy to use correction factors.

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[1] TRANSLATOR'S NOTE. The table has been transposed without much alteration from the German edition. The data may not therefore apply in all cases to American conditions, and this is especially true of the first costs assumed and the wage accounts. The latter have been corrected by assuming an average wage scale of 20 cents per hour, but the items referring to cost of oil, waste, costs of fuel, etc., have not been changed since the entire set of figures is hypothetical and is subject to large variations.

TABLE 40

FIRST COSTS AND COSTS OF OPERATION, ILLUMINATING-GAS AND SUCTION-GAS INSTALLATIONS

BASED UPON 1 B.H.P. OR 1 B.H.P.-HOUR RESPECTIVELY, AND ASSUMING 3000 WORKING HOURS PER YEAR

Figures for Illuminating-gas Engines in *Italics*.

	Nominal Horse-power N_n	5	10	15	20	25	30	35	40	50	60	70	80	100	125	150	175	200
	I. First Costs of Engines and Producers, including Auxiliaries, per B.H.P.																	
1	Engine only, including erection ..... \$	*97.5*	*89.0*	*82.5*	*77.5*	*74.0*	*70.0*	*67.5*	*66.5*	*64.0*	*62.5*	*62.0*	*61.5*	*59.5*	*57.5*	*56.5*	*55.0*	*54.0*
2	Producer only, including erection ..... \$	60.0	47.5	37.5	31.2	25.0	21.2	18.7	16.2	14.5	13.7	13.0	12.0	11.2	10.5	10.0	9.5	8.7
3	Engine auxiliaries (belt pulley, piping, starting gear, etc.) ..... \$	*8.7*	*7.0*	*6.0*	*5.2*	*4.5*	*5.5*	*5.4*	*5.2*	*4.9*	*4.5*	*4.2*	*4.0*	*3.6*	*3.4*	*3.2*	*3.1*	*3.0*
4	Same for the suction-gas engine installation . \$	12.5	11.2	9.5	8.2	7.2	8.7	8.5	8.2	7.7	7.2	6.7	6.2	5.7	5.4	5.2	5.1	5.0
5	Total cost $\left(\frac{A_m}{N_n}\right)$, illuminating-gas engine ready for operation ..... \$	*106.2*	*96.0*	*88.5*	*82.7*	*78.5*	*75.5*	*72.9*	*71.7*	*68.9*	*67.0*	*66.2*	*65.5*	*63.1*	*60.9*	*59.7*	*58.1*	*57.0*
6	Same for the engine and producer installation ..... \$	170.0	147.7	129.5	116.9	106.2	99.9	94.7	90.9	86.2	83.4	81.7	79.7	76.4	73.4	71.7	69.6	67.7
	II. Operating Costs due to the Sum $\frac{A_m}{N_n}$, per B.H.P.-hour. (Interest 4%, Depr. 8%, total 12%.)																	
7	On total cost of illuminating-gas installation (line 5) ..... cts.	*.425*	*.386*	*.360*	*.330*	*.306*	*.302*	*.292*	*.285*	*.275*	*.267*	*.265*	*.260*	*.250*	*.242*	*.237*	*.232*	*.227*
8	On total cost of suction-gas installation (line 6) ..... cts.	.680	.590	.517	.467	.425	.400	.380	.362	.345	.335	.327	.317	.305	.292	.285	.277	.270
	III. First Costs of Buildings and Foundations. (Price assumed for building, \$1.15 per sq.ft.) (Price assumed for foundations, \$0.183 per cu.ft.)																	
9	Floor space for ill.-gas engine only ..... sq.ft.	*19.4*	*15.1*	*13.5*	*11.9*	*9.7*	*8.6*	*8.1*	*7.6*	*7.0*	*6.5*	*5.9*	*5.3*	*4.9*	*4.6*	*4.4*	*4.2*	*4.1*
10	Foundation for ill.-gas engine only ..... cu.ft.	*14*	*14*	*14*	*14*	*14*	*14*	*14*	*14*	*14*	*14*	*14*	*14*	*14*	*14*	*14*	*14*	*14*
11	Floor space for suction-gas engine and producer ..... sq.ft.	43.0	31.2	25.4	21.0	17.2	15.1	14.0	12.9	11.8	10.8	9.8	9.0	8.2	7.7	7.3	7.1	6.9
12	Foundation for suction-gas engine and producer ..... cu.ft.	17.7	17.7	17.7	17.7	17.7	17.7	17.7	17.7	17.7	17.7	17.7	17.7	17.7	17.7	17.7	17.7	17.7
13	Total cost $\left(\frac{A_g}{N_n}\right)$ of building and foundation for ill.-gas engine inst. (from lines 9 and 10) . \$	*25.0*	*20.0*	*18.1*	*16.2*	*13.7*	*12.5*	*11.9*	*11.2*	*10.6*	*10.0*	*9.4*	*8.7*	*8.2*	*7.9*	*7.6*	*7.5*	*7.4*
14	The same for the suction-gas inst. (from lines 11 and 12) ..... \$	53.2	39.4	32.5	27.5	23.1	20.6	19.4	18.1	16.9	15.6	14.5	13.5	12.6	12.0	11.5	11.4	11.1

	Nominal Horse-power N_n	5	10	15	20	25	30	35	40	50	60	70	80	100	125	150	175	200
	IV. Operating Costs due to the Sum $\frac{A_g}{N_n}$ per B.H.P.-hour. (Interest 4%, Depr. 2%, total 6%.)																	
15	On total building cost of ill.-gas engine (line 13) . . . cts.	*.050*	*.040*	*.034*	*.032*	*.027*	*.025*	*.024*	*.022*	*.021*	*.020*	*.019*	*.018*	*.017*	*.016*	*.015*	*.015*	*.015*
16	On total building cost of suction-gas inst. (line 14) . . . cts.	.106	.079	.065	.055	.047	.041	.039	.036	.034	.031	.029	.028	.027	.024	.023	.023	.022
	V. Incidental Expenses per B.H.P.-hour. (Attendance, Maintenance and Repairs, Lubrication, etc.)																	
17	Cost of attendance, ill.-gas engine only [eq. (5)] . . . cts.	*.223*	*.158*	*.129*	*.112*	*.100*	*.092*	*.085*	*.079*	*.071*	*.065*	*.060*	*.056*	*.050*	*.045*	*.041*	*.038*	*.035*
18	Cost of attendance, suction-gas installation [eq. (5a)] . . . cts.	1.115	.790	.645	.560	.500	.460	.425	.395	.355	.325	.300	.280	.250	.225	.205	.190	.175
19	Maintenance, ill.-gas inst., including building, etc. . . . cts.	*.131*	*.115*	*.107*	*.100*	*.092*	*.087*	*.085*	*.082*	*.080*	*.077*	*.075*	*.074*	*.071*	*.068*	*.067*	*.065*	*.062*
20	Maintenance, suction-gas installation, including building, etc. . . . cts.	.223	.187	.162	.145	.130	.120	.115	.109	.102	.099	.096	.093	.089	.085	.083	.081	.079
21	Cost of oil and waste, ill.-gas engine . . . cts.	*.120*	*.100*	*.098*	*.097*	*.095*	*.095*	*.093*	*.093*	*.090*	*.090*	*.087*	*.087*	*.085*	*.082*	*.080*	*.078*	*.075*
22	Cost of oil and waste, suction-gas inst. . . . cts.	.150	.125	.120	.120	.115	.115	.112	.112	.110	.110	.108	.108	.105	.102	.100	.098	.095
	VI. Total Incidental Expenses per B.H.P.-hour.																	
23	Illuminating-gas engine installation . . . cts.	*.949*	*.799*	*.728*	*.671*	*.620*	*.601*	*.579*	*.561*	*.537*	*.519*	*.506*	*.495*	*.473*	*.453*	*.440*	*.428*	*.414*
24	Suction-gas installation . . . cts.	2.274	1.771	1.509	1.347	1.217	1.136	1.071	1.014	.946	.900	.860	.826	.776	.728	.696	.669	.641
	VII. Cost of Fuel, per B.H.P.-hour.																	
25	Consumption of ill. gas per B.H.P.-hour . . cu.ft.	21.2	19.3	19.1	18.7	18.7	18.4	18.4	18.0	17.7	17.3	16.9	16.6	16.2	15.9	15.5	15.2	14.8
26	Consumption of anthracite per B.H.P.-hour lbs.	1.47	1.32	1.25	1.21	1.10	1.10	.99	.99	.95	.95	.92	.92	.90	.88	.86	.84	.83
27	Cost of ill. gas (at 85c. per 1000 cu.ft.) . . . cts.	1.80	1.65	1.62	1.60	1.60	1.55	1.55	1.52	1.50	1.48	1.44	1.41	1.38	1.35	1.32	1.29	1.26
28	Cost of coal (at $6.80 per ton) . . . cts.	.50	.45	.43	.41	.38	.38	.34	.34	.33	.33	.32	.32	.31	.30	.29	.28	.28
	VIII. Total Operating Costs per B.H.P hour. (Consisting of Items VI and VII.)																	
29	Illuminating-gas engine installation . . . cts.	**2.749**	**2.449**	**2.348**	**2.271**	**2.220**	**2.151**	**2.129**	**2.081**	**2.037**	**1.999**	**1.946**	**1.905**	**1.853**	**1.803**	**1.760**	**1.718**	**1.674**
29*a*	Of this amount the incidental expenses (Items II, IV, V, and VI) constitute . . . %	34.0	32.6	31.0	29.6	28.0	27.8	27.2	27.0	26.3	26.0	26.0	25.9	25.6	25.2	25.0	24.9	24.8
30	Suction-gas installation . . . cts.	**2.774**	**2.221**	**1.939**	**1.757**	**1.597**	**1.516**	**1.411**	**1.354**	**1.276**	**1.230**	**1.180**	**1.146**	**1.086**	**1.028**	**.986**	**.949**	**.921**
30*a*	Of this amount the incidental expenses (Items II, IV, V, and VI) constitute . . . %	82.	80.0	78.0	77.0	76.5	75.0	75.5	75.0	74.2	73.3	72.8	71.7	71.0	70.8	70.5	70.5	69.6

NOTE. Items 1 and 2: Cost of erection, etc., taken at approximately 3% of purchase price for illuminating-gas engine installation, and at approximately 5% for suction-gas plant. Items 3 and 4: It has been assumed that up to 25 H.P. simple starting cranks are used; above that power, allowance is made for air compressor and auxiliaries. Cost of piping and other accessories have been taken at from 6-8% of the cost of the rest of the equipment. Items 10 and 12: The size of the foundation seems to vary without regard to horse-power, and may be anything from 7 to 35 cu.ft per B.H.P. For the ordinary type of horizontal engine without out-board bearing, an allowance of from 12-17 cu.ft per B.H.P. is ample, providing that the soil is safe.

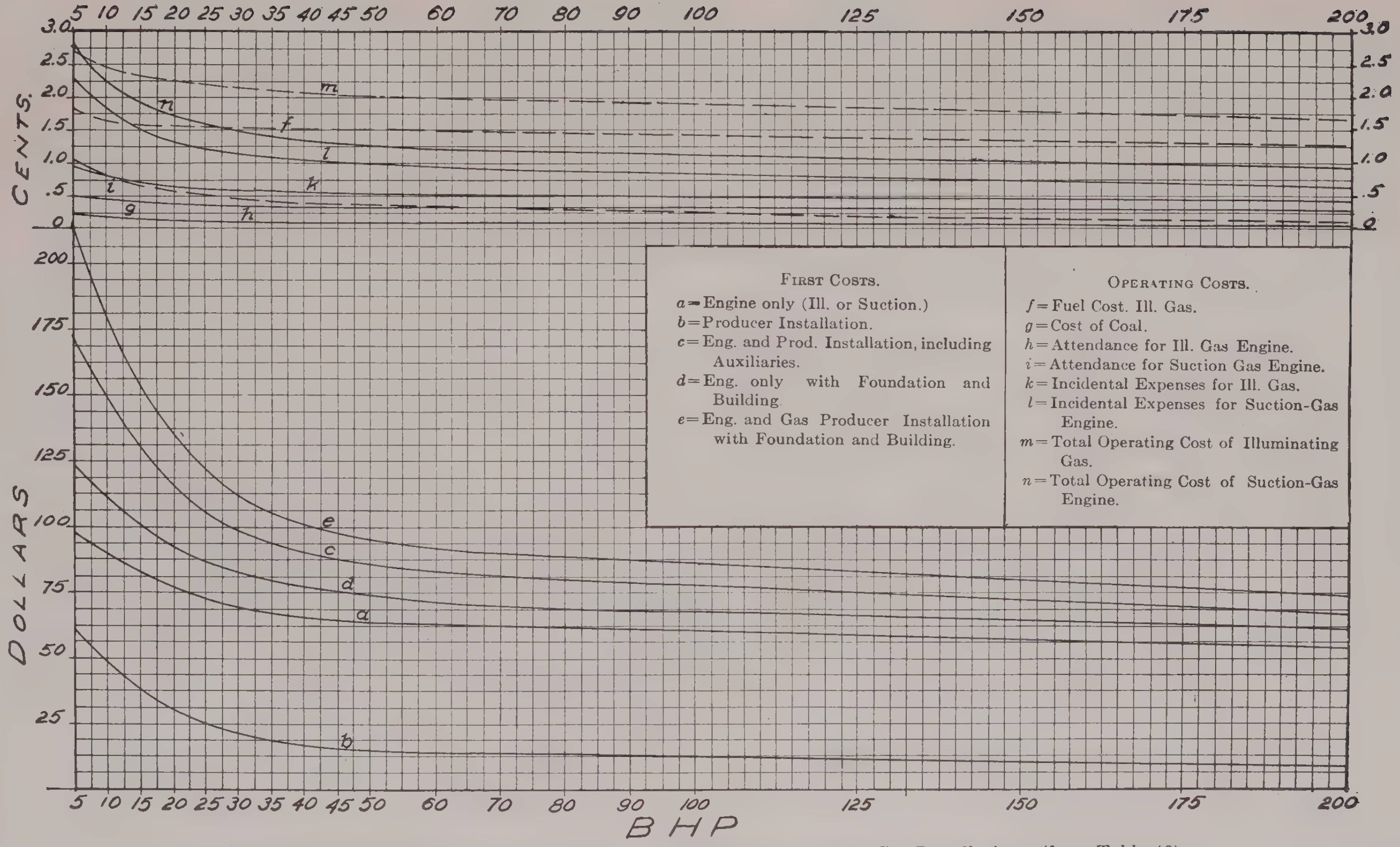

FIG. 470.—First Costs and Operating Costs of Illuminating and Suction-Gas Installations, (from Table 40).

A close study of the diagram, Fig. 470, brings out many things that are not so apparent from a survey of the table. Among these may be mentioned the following:

(*a*) Rapid increase of both first cost and operating costs with decrease in rated horse-power. (Unfavorable to the operation of small plants.)

(*b*) The small ratio of incidental expenses (curves **h** and **k**) to total operating costs (curve **m**) in the case of the illuminating-gas plants (points to the latter as well suited for small capacity and intermittent operation).

(*c*) The great preponderance of the incidental expenses (curve **l**) over the fuel cost (curve **g**) in the case of the suction-gas plant.

(*d*) The rapid increase of the first cost of gas producers (curve **b** as compared with the total first cost of the entire installation (curve **e**), for capacities less than 30 B.H.P.

(*e*) The approach of the curves **m** and **n**, showing total operating costs for illuminating-gas and suction-gas plants respectively, at the left-hand side of the diagram.

(*f*) The considerable proportion that the cost of buildings and foundations bears to the total first cost of horizontal machines (distance between curves **c** and **e**, or **a** and **d**).

In most comparisons of operating costs, hydraulic power is generally, without much consideration, assumed to be the cheapest. As a matter of fact, however, this assumption is justified only in cases where this power can be made available with simple means and at no great expense, and where the supply of water is uniform enough to admit of the elimination of costly reserve machinery. Where these conditions do not obtain, it may be found that the first cost of the installation, and the operating costs resulting therefrom are considerably higher than the same items for gas or steam plants. As an example of this fact, the following figures were taken from a report on a 1000-H.P. turbine installation:[1]

Quantity of water available, 141 cu.ft. per sec., under a head of 8.20 ft.

3 turbines, each 350 H.P. cost.	$18750
3 relay steam engines and boilers, cost	60000
Total cost, including generators, buildings, etc.	275000
First cost per B.H.P., at turbine shafts	140.00
First cost per B.H.P., at switchboard	212.50
Operating cost per B.H.P. hour, based on general output	1.35 cents

The question of the relative superiority of steam and gas as motive power has led to a " war of words " between the manufacturers, which has been merely intensified by the introduction of the suction-gas plant. With regard to this controversy, the author desires to simply note the following:

In conjunction with good gas producers, and assuming average fuel prices, the gas engine is with certainty able to compete with steam plants up to 4 or 500 H.P. With cheap gas (less than 70 cents per 1000 cu.ft.) the illuminating-gas engine is economically superior to the steam engine up to 50 H.P. Stationary oil engines usually enter the field when illuminating gas cannot be had. An exception to this is the Diesel engine (and possibly also the Haselwander engine) which, if cheap oils (less than 10 cents per gallon) are available, remains a serious competitor to steam also for the larger sizes and which, where the operating hours are short or where there are frequent interruptions, may even be preferable to a suction-gas installation.

In general, gas is economically superior to steam whenever the power demand is fluctuating or intermittent and where rapid and easy erection as well as clean operation without molestation of any kind are a consideration. Of course where steam power cannot be used on account of government restrictions or other reasons, or where on this account its use is rendered difficult, gas power is a very welcome substitute.

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[1] Süddeutsche Bauzeitung, 1905, p. 74.

In some of the special types of construction, as direct-connected units for instance, the ability of the internal-combustion engine to compete with the steam engine is even more pronounced than in the case of simple mill engines, but local conditions have a great influence in this respect. The following figures, however, can be obtained with certainty, provided the installation is in good shape and the conditions are normal.

In direct-connected generator units:

30 to 26.5 cu.ft. of illuminating gas per K.W. hour;
2.6 to 1.9 lbs. of anthracite per K.W. hour;
3.3 to 2.6 lbs. of coke per K.W. hour;
or 1 cu.ft. of illuminating gas will generate from 33.5 to 38.0 watt hours;
1 lb. of anthracite will generate from 380 to 530 watt hours;
and 1 lb. of coke will generate from 300 to 390 watt hours.

In pumping engines:

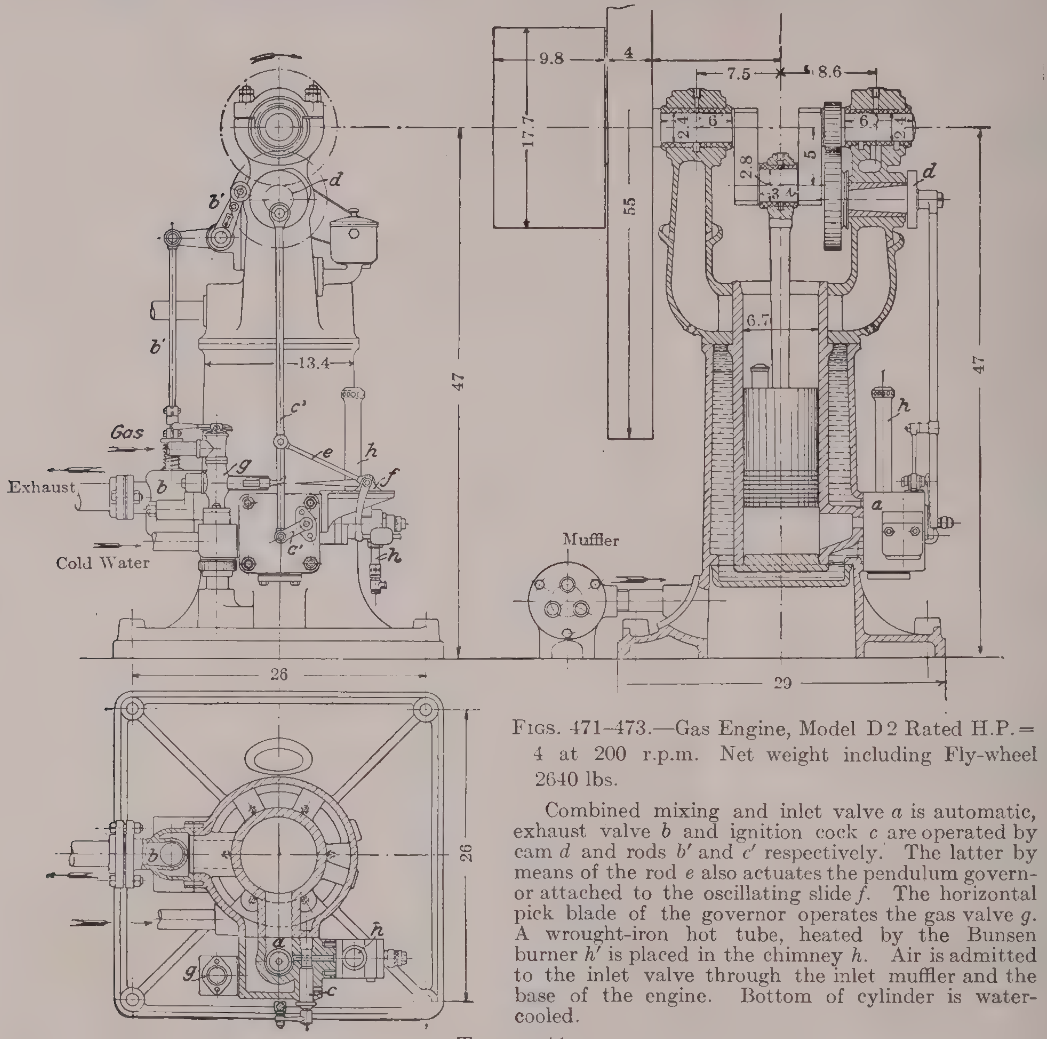
$$\left.\begin{array}{l}\text{1 cu.ft. of illuminating gas will show a duty of from 72–83 000 ft.-lbs.}\\ \text{1 lb. of coke will show a duty of from 1 000 000–1 250 000 ft.-lbs.}\\ \text{1 gal. gasoline will show a duty of from 7 775 000–8 000 000 ft.-lbs.}\end{array}\right\}\text{in water lifted.}$$

These values are obtained when the load factor is high, that is, when the plant is operated near normal rating, and refer to standard conditions of fuel. Losses through starting, stand-by losses, and belt losses have been taken into account. In making guarantees the very strong influence that the load factor has upon plant economy must not be left out of consideration. No matter how carefully the plant may otherwise be constructed and operated, leaving out of the computation the actual average load on the installation may result in complete failure in meeting the guarantees made. If therefore the fuel consumption only is the deciding factor in the acceptance, the guarantee should be based upon the consumption at or near normal rating.

In acceptance trials or other tests it frequently happens that differences of opinion crop up as to the allowances that should be made for unavoidable fuel losses (fuel used until all conditions become constant, fuel not burned at end of trial, efficiency of transmission of power consumers, etc.). For that reason it is well to have all the conditions thoroughly understood at the time of purchase, or at the very latest just before the beginning of the trial. As far as the engine builder is concerned, the thing of most importance to him is probably the accurate determination of the efficiency of the power consumers to be operated (generators, pumps, blowing cylinders, etc.), especially since the manufacturers of these machines are very prone to give out these efficiencies more or less too high. The power loss in belting also is in many acceptance trials a factor of importance. For lack of a better way, this loss is generally estimated, and it is consequently found that in plants of approximately equal capacity the allowances vary between 2 and 10% of the engine capacity. In reality this loss is probably on the average 4 or 5%; in one definite case, however, tests made in the laboratory of a German technical school on a 20 H.P. engine showed this loss to be 10% of the total power transmitted. In view of such uncertainties, to say nothing of the unavoidable errors of observation, the leeway of 5% which is commonly allowed on all guarantee figures is certainly justified.

II. Types of German Engines

1. Gasmotoren-Fabrik Deutz in Cöln-Deutz. (See also Plates VIII to XI.)



Figs. 471–473.—Gas Engine, Model D 2 Rated H.P. = 4 at 200 r.p.m. Net weight including Fly-wheel 2640 lbs.

Combined mixing and inlet valve *a* is automatic, exhaust valve *b* and ignition cock *c* are operated by cam *d* and rods *b'* and *c'* respectively. The latter by means of the rod *e* also actuates the pendulum governor attached to the oscillating slide *f*. The horizontal pick blade of the governor operates the gas valve *g*. A wrought-iron hot tube, heated by the Bunsen burner *h'* is placed in the chimney *h*. Air is admitted to the inlet valve through the inlet muffler and the base of the engine. Bottom of cylinder is water-cooled.

Table 41

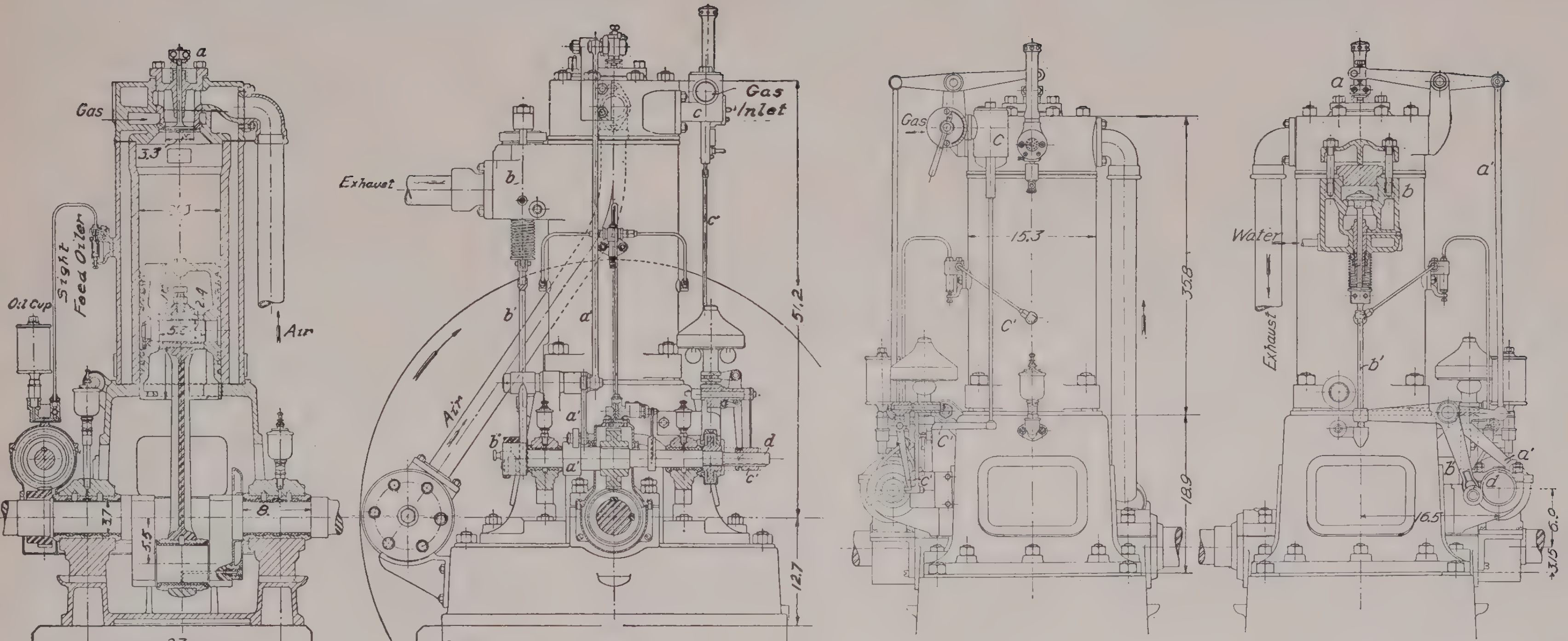
DIMENSIONS AND WEIGHTS OF MODEL D 2 ENGINES

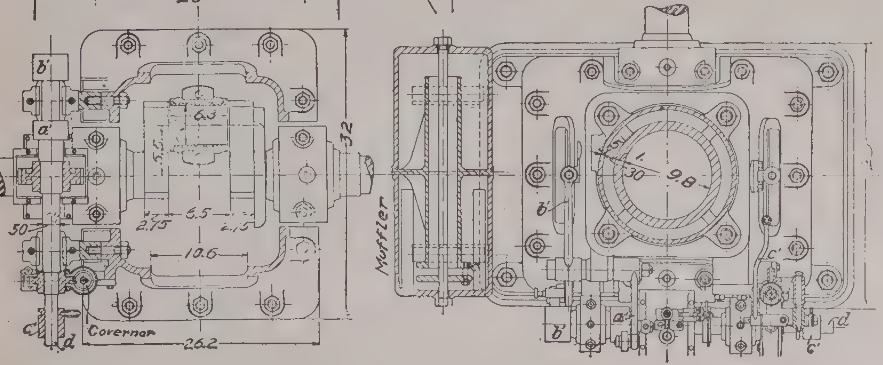
N_n in B.H.P.	½	1	2	3	4
R.P.M.	240	230	210	210	200
Length* ft.	2.44	3.20	3.50	3.80	4.27
Width† "	2.01	2.38	2.53	2.84	3.17
Height‡ "	3.20	3.50	3.60	3.66	3.66
Net weight lbs.	750	1080	1500	1930	2640
Total weight "	965	1430	1830	2330	3120

* Across the shaft, therefore equal to fly-wheel diameter.

† Along the shaft.

‡ To center of shaft. Total height is greater by one-half wheel diameter.





FIGS. 474–479.—Gas and Oil Engine, Model H 3. Rated B.H.P. = 16 at 300 r.p.m. Net weight including Fly-wheel 4840 lbs.

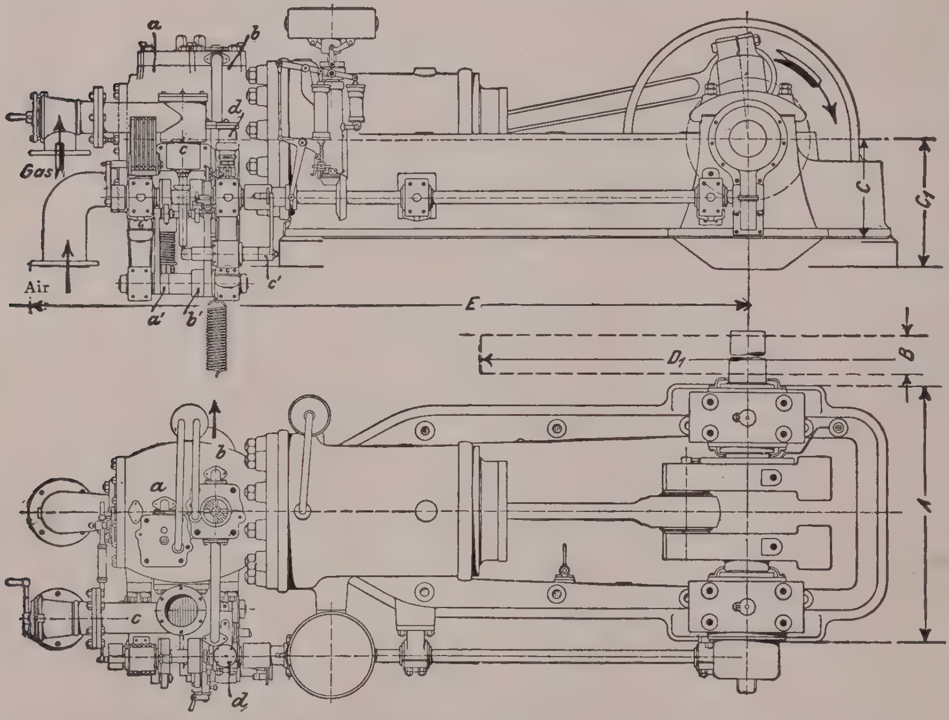
Inlet valve *a*, exhaust valve *b*, and gas valve *c* are operated by separate cams on the lay shaft *d* through levers or rods *a′*, *b′*, and *c′* respectively. The gas cam *c′* is cone-shaped and under the action of the centrifugal governor is movable along shaft *d*. Regulation is thus effected by controlling the quality of the mixture. Ignition is by simple open hot tube. All of the sizes are furnished with two overhanging fly-wheels, and may be made to operate on either illuminating gas or oil.

TABLE 42

DIMENSIONS AND WEIGHTS OF MODEL H 3 ENGINES

N_n in B.H.P.	6	8	10	12	16
R.P.M.	300	300	300	300	300
Length, ft.	4.27	4.35	4.76	4.76	5.09
Width, "	4.30	4.87	5.42	5.42	5.88
Height, "	5.75	6.20	6.24	6.24	6.86
Net weight*, lbs.	2530	3400	4050	4290	4840
Shipping weight, "	3080	4380	4840	5170	5720

* These and all the weights given in what follows refer to mill engines. For electric-lighting service ($\delta = \frac{1}{70}$) the engines are heavier by from 15–20%.



FIGS. 480 and 481.—Gas Engine (Old Type G 6 III) for Capacities from 35–150 B.H.P.

Both inlet valve *a* and exhaust valve *b* are placed on the under side of the cylinder head; *c* is the gas valve, *d* the starting valve. All valves are operated by cams, the valve levers are marked *a'*, *b'*, and *c'* respectively. The make-and-break ignition is placed in the axis of the cylinder. Speed is regulated by hit-and-miss or quality governing. For capacities from 50 to 150 H.P., the firm has lately introduced a new model G 9 which, as far as frame and cylinder are concerned, is the same as G 6 III, but has the arrangement shown in Plates IX and X regarding valves, valve gear, and governing. The dimensions given in the following table will serve approximately for both types:

TABLE 43

DIMENSIONS AND WEIGHTS OF MODEL G 6 III ENGINES

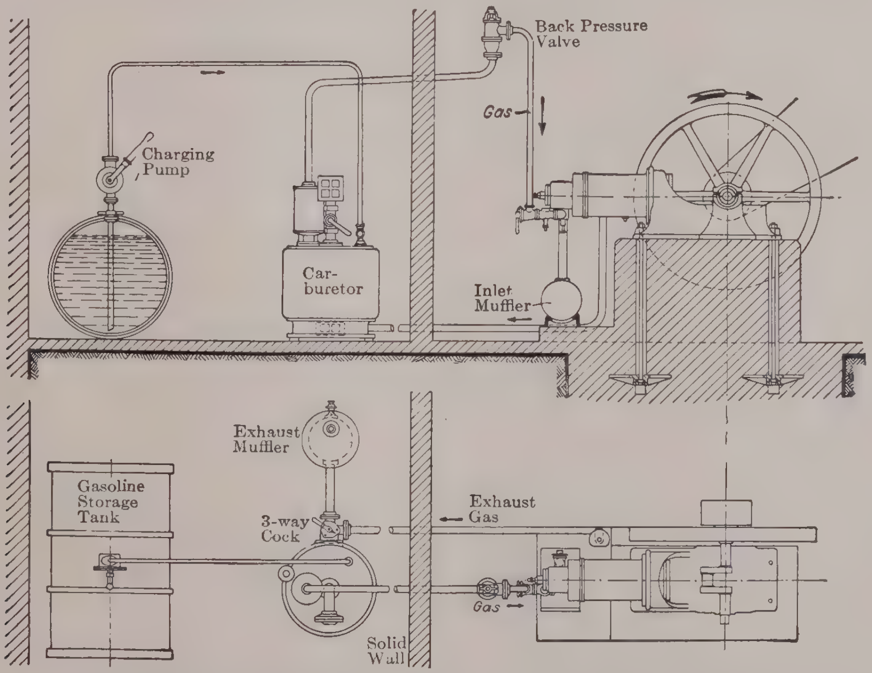
N_n in B.H.P.		35	40	50	60	70	80	100	125	150
R.P.M.		190	190	190	190	180	180	170	160	140
Dimensions in feet	A	3.70	3.70	4.14	4.19	4.72	4.72	5.43	5.96	7.17
	B *	.79	.82	.95	1.03	1.15	1.21	1.31	1.47	1.80
	C	1.39	1.39	1.56	1.56	1.64	1.64	1.80	1.97	2.38
	C_1	2.62	2.62	2.79	2.79	2.95	2.95	3.12	3.28	3.61
	D_1†	7.22	7.55	7.88	8.20	9.18	9.18	9.52	10.30	11.50
	E	10.35	10.35	11.30	11.30	11.85	11.85	14.00	14.95	16.15
Net weight	lbs.	15400	19800	21 200	24200	28 600	30800	35 200	.....	.....

Up to 80 B.H.P. inclusive, with or without outboard bearing; in the latter case there are two fly-wheels.

* For mill and electric lighting purposes.

† For mill purposes.

Since the beginning of 1904 the Deutz Co. use the double-acting 4-cycle engine for all powers above 150 B.H.P. The main features of the design of this machine are shown in Plates IX and X. Greater capacities are obtained by the combination of cylinders side by side or tandem, or both, resulting in twin or tandem, or twin-tandem engines. In this way capacities up to 6000 B.H.P. may be reached.



FIGS. 482 and 483.—Erecting Plan for a Deutz Gasoline Engine, Model E.

The room for gasoline storage is separated from the engine room by a solid wall (following the insurance regulations required by the Union of German Fire Insurance Societies). The gasoline tank should never be located higher than the carbureter to prevent syphoning.

Operating Results. (*a*) *Illuminating Gas Engine,* horizontal cylinder; 50 B.H.P.; $D=15''$, $S=22.85''$; ratio of compression $\varepsilon=5.92$; ignition by hot tube; speed regulation by changing quality of mixture. Tests made by Prof. E. Meyer,[1] August, 1908, showed the following results:

TABLE 44

Test No.	R.P.M. n	B.H.P. N_e	I.H.P. Total, N_{i+}	I.H.P. Net,* N_i	Mean Effective Pressure, from N_i p_i lbs. per sq.in.	$N_r = N_i - N_e$ H.P.	Mean Friction Resistance, p_r lbs. per sq.in.	$\eta_m = N_e \div N_i$ %	Gas Consumption per B.H.P.-hour. cu.ft.	Gas Consumption per I.H.P.-hour. cu.ft.	Heat Lost in Cooling Water per Cu.ft. of Gas. B.T.U.	Exhaust Temp. Deg. F.
1	198.8	64.2	77.0	72.4	70.5	8.2	8.10	89	15.65	13.90	210.15	929
2	204.6	53.7	67.2	62.5	59.2	8.8	7.95	86	17.75	15.40	199.00	850
3	206.3	42.0	56.5	51.8	48.7	9.8	9.25	81	20.45	16.60	194.50	738
4	207.9	30.3	44.5	39.5	36.8	9.2	8.67	76	28.90	21.10	186.50	616
5	206.0	17.7	33.8	29.0	27.3	11.3	10.70	61	44.10	26.90	170.00	435
6	200.6	0	17.0	12.3	11.9	12.3	12.00	0	417.0 per hour	33.90	......	287

* See footnote p. 342.

The gas consumption is referred to 32° F., 760 mm. barometer, and a heating value of 560 B.T.U. per cu.ft.

The compression pressure was 131 lbs. and the explosion pressure at maximum load from 312 to 355 lbs. per sq.in. Diagrams obtained at maximum load are shown in Fig. 484. The rounded tip of the combustion line indicated strong after-burning up to 1/15 of the expansion stroke.

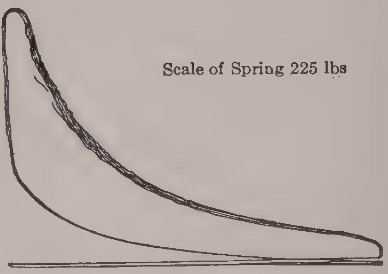


FIG. 484.

(*b*) In 1895, O. Körner obtained the following values with two of the smaller horizontal machines:

Normal Rating, B.H.P.		8	25
R.P.M.		222	201.2
B.H.P.		9.86	29.70
Heating Value of Gas, B.T.U. per cu.-ft.		566	560
Gas Consumption, cu.ft. per B.H.P.-hour	at 32° and 760 mm.	18.50	17.20
Gas Consumption, at no load, cu.-ft. per hour	including ignit. flame	36.50	91.50

(*c*) *Pressure Producer Gas Engine,* 160 B.H.P., 2 cylinder opposed, each of dia. 20.5″, stroke 30″, ratio of compression $\varepsilon = 4.85$. Electric ignition, hit-and-miss regulation. Each cylinder has two exhaust valves, the discs of which are cooled by spraying from below with water. Tests were conducted by E. Meyer in April, 1896, on an engine in a water works.[2]

The compression pressure was 115.5 lbs., the explosion pressure, on account of slow combustion, was but little higher than this. Heating value of the coke 12,995 B.T.U. per lb., of the producer gas was on the average 135 B.T.U. per cu.ft., referred to 32° F. and 760 mm. barometer.

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[1] Z. d. V. D. I., 1900, p. 303.

[2] Z. d. V. D. I., 1896, p. 1241.

TABLE 45

R.p.m.	Mean Effective Pressure, p_i	I.H.P.	H.P. in Water Lifted,	B.H.P. Computed,	Coke Consumption on Basis of						Cooling Water per I.H.P.-hour.	Water Vapor per Pound Producer Coke.
					N_i		N_w		N_e			
					Producer. lbs.	Boiler. lbs.	Producer, lbs.	Boiler. lbs.	Producer. lbs.	Boiler. lbs.		
n	lbs. per sq.in.	N_i	N_w	N_e							cu.ft.	lbs.
141.4	48.00	169.0	121.5	142.0	1.27	.132	1.785	.193	1.51	.152	.61	.614
					1.402		1.978		1.662			
137.75	51.20	176.0	130.0	149.6	1.28	1.42	1.735	.193	1.49	.152	.81	.642
					1.422		1.928		1.642			

One of the older types of pressure producer gas engines of 100 H.P., tested in 1897 by O. Köhler, gave the following figures:

Fuel used in the producer, German anthracite, heating value 14400 B.T.U. per lb.
Fuel used in the boiler, gas coke.
Mean heating value of gas at 32° F. and 760 mm. bar., 146 B.T.U. per cu.ft.
Mean R.P.M., 160.93, B.H.P. 114.1.

Fuel consumption per B.H.P., per hour	in producer....	.966 lbs.
" " " " " "	in boiler......	.126 "
" " " " " "	total..........	1.092 "

(d) *Results of Tests on several Deutz Suction Gas Installations.* Test by F. Kapeller, Director of the Industrial School, Nürnberg, 1902.

Engine rating, 20 B.H.P.
B.H.P. on test, 23.10.
Consumption of anthracite per B.H.P.-hour, .995 lbs.
Coal, from Langenbrahm mine, had a heating value of 14940 B.T.U. per lb., and contained 2.71% of ash.

At about the same time tests made by A. Stauss in the mechanical laboratory of the technical school at Stuttgart gave the following results:[1]

Normal rating, 70 H.P.
Cylinder diameter, 16.10″
Stroke, 23.6″.
R.p.m., approx., 182.
Mean piston speed, 11.95 ft. per sec.
Compression press., 9 at. = 128 lbs. per sq.in.

Langenbrahm anthracite, size $\frac{3}{8}$–$\frac{5}{8}$″.
Heating value of anthracite, 13 925 B.T.U. per lb.
Composition of coal:
C = 84.60%, H = 3.49%
O + N = 3.49%, S = .92%
Ash = 5.82%, Water = 1.20%

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[1] See Journal für Gasbeleuchtung, 1902, Heft 29, and following.

Time of reading	11:30–2:44	2:44–6:48
Mean effective pressure	68.0	66.6
I.H.P.	72.25	71.60
B.H.P.	64.10	63.90
Mechanical efficiency, %	88.65	89.35
Consumption of anthracite per I.H.P. hour, lbs. net	.897	.787
Consumption of anthracite per B.H.P. hour, lbs. net	1.010	.892
Consumption of water (steam) per I.H.P. hour, lbs.	0.492	.494
Consumption of water (steam) per B.H.P. hour, lbs.	.556	.550
Water per lb. of anthracite, lbs.	1.205	1.372
Water per I.H.P. hour, lbs.	16.25	16.48
Water per B.H.P. hour, lbs.	18.35	18.55
Cooling water per I.H.P. hour, lbs.	43.10	32.90
Cooling water per B.H.P. hour, lbs.	48.70	36.70
Total water consumption per I.H.P. hour, lbs.	59.35	49.38
Total water consumption per B.H.P. hour, lbs.	67.05	55.25
Outlet temperature of scrubber water, ° F.	117.8	109.2
Outlet temperature of cooling water, ° F.	136.4	154.4
Efficiency of producer, %	86.36	86.82
Economic efficiency of entire plant, %	17.99	20.61
Heat lost in scrubber, %	8.84	8.93
Heat lost in cooling water, %	29.79	31.14
Heat lost in exhaust, %	36.28	32.61
Composition of producer gas, volume %:		
CO	22.0	23.3
H_2	17.0	17.4
CH_4	1.8	2.0
C_nH_n	0	0
CO_2	5.8	5.5
O_2	.7	.5
N	52.7	51.3

Composition of the exhaust gases:

$CO_2=5.6\%$, $O_2=13.6\%$, $N=80.5\%$, combustible (taken as CO) $= .3\%$. Pressure in gas mains: 3–5″ water below atmosphere.

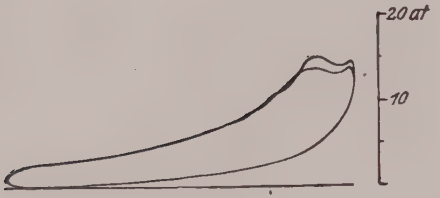

Fig. 485.

The explosion pressures varied between 200 and 230 lbs per sq.in. In all of the tests the combustion was so slow that the diagrams, see Fig. 485, closely resembled those of a constant pressure engine (Diesel for example).

Tests made by R. Mathot, March, 1904, on a 60 H.P. engine having 16.5″ cylinder diameter and 18.9″ stroke, gave the following main results:

Load	Full.	Half.	No Load.
Duration of tests, hours	4.5	4.5	—
Revolutions per minute	188.66	195.5	199.0
Mean effective pressure, pounds per square inch	73.0	44.3	11.8
I.H.P.	75.9	44.35	10.83
B.H.P.	64.20	33.35	—
Mechanical efficiency, %	84.6	75.0	—
Compression pressure, pounds per square inch	170.0	121.0	92.3
Explosion pressure, pounds per square inch	383.0	249.0	213.0
Pressure in exhaust, pounds per square inch	24.1	19.8	0
Maximum suction pressure, pounds per square inch	−4.26	−6.52	−8.52
Relative increase of speed (full load = 1.0)	1.0	1.035	1.052
Consumption of anthracite per I.H.P. hour, lbs. net	.68	.876	—
Consumption of anthracite per B.H.P. hour, lbs. net	.798	1.170	—
Economic efficiency of entire plant, %	24.3	16.0	—

The coal used came from the Mossbach mine near Aix-la-Chapelle, size $\frac{5}{8}$–$\frac{3}{4}$″, ash 7.33%, water 2.69%, heating value 13638 B.T.U. per lb. The gas had a heating value of 140 B.T.U. per cu.ft. At full load, the outlet temperature of the jacket water from the cylinder head was 109.4°, from the cylinder barrel was 127.4°, while the temperature of the vaporizer was 183.2°.

The following figures are taken from the results of a two-day test made by Prof. Witz and Mr. Mathot in 1894 on a 200 H.P. double acting 4-cycle engine ($D=21.2''$, $S=27.5''$, piston rods 4.72 and 4.33″).

	First Day.	Second Day.
Duration of test, hours	3	10
Average revolutions per minute	151.29	150.20
B.H.P.	211.8	220.00
Consumption of anthracite per B.H.P. hour, lbs.	.937	.726
Economic efficiency, entire plant, %	19.0	24.4
Total water consumption per B.H.P. hour, lbs.	—	77.8
Of this amount there were used in cylinder jackets	—	45.8
in piston and rod	—	17.25
in producer vaporizer	—	.65
in scrubber	—	14.10
Water vapor per pound of anthracite, lbs.	—	1.92

The temperature of the cooling water was 117.8° leaving the piston and 134.9° leaving the cylinder jackets. The gas had a temperature of 557.6° leaving the producer and 62.6° after passing the scrubber. The kind of anthracite used (Bonne Esperânce et Batterie, Herstal) had a heating value of 14580 B.T.U. per lb. When banked over night the producer burned 47 lbs. of coal, which is approximately 10% of the daily consumption.

(*e*) The following table shows average results obtained on capacity tests made by the firm on their new *brown-coal installation.* The plant was one of the first ones constructed and developed 60 H.P.

TABLE 46

		Rhenish Brown Coal.			Styrian Brown Coal.
		Test I	Test II	Test III	
Composition of coal as mined	C	23.86 %	30.99 %	29.78%	44.22%
	H	1.82	2.10	2.03	3.58
	O+N	9.21	11.44	11.975	17.74 + .52
	S*	.29	.25	.085	.29
	Ash	4.64	2.67	2.18	5.86
	Water	60.18	52.55	53.95	27.79
		100.00	100.00	100.00	100.00
Combustible in coal, %		35.18	44.78	43.87	66.35
Heating value of coal, B.T.U. per pound		3283	4320	4257	6840
Average composition of gas	CO_2	8–9%	8%	7–8%	3.85%
	CO	23–25	25	27	28.70
	H	12–13	15	15	13.65
Combustible in gas, %		35–38	40	40	42.35
Average heating value of gas, B.T.U. per cubic foot		123	140	140	146
Gas obtained per pound of coal, cu.ft.		18.1	25	24.5	35.3
Gas consumed per B.H.P. hour, cu.ft.		93–96.5	86.2	82.1	about 78.5
Coal consumed per B.H.P. hour, lbs.		5.18	3.35	3.35	“ 2.23

* Exclusive of sulphur in ash.

It is stated that these brown-coal producers are capable of gasifying very inferior coals containing as high as 60% of water, and that they may be banked for periods even exceeding 14 days. The starting up after such periods, it is claimed, does not take over 15 minutes.

Tests were also made with air-dried peat (16.57% water); the peat consumption per B.H.P. hour is given as 2.84 lbs.

One of the new double-generator plants for brown coal (see Fig. 432, p. 283) was tested by Prof. E. Meyer in June, 1903, with the following results:

Test No.	I	II	III
Duration of test, hours	9½	12	10
Revolutions per minute	181.8	181.6	182.7
Brake H.P. $=N_e$	69.50	69.45	no load
Positive ind. mean pressure, p_i+ lbs. per sq.in.	67.1	67.5	27.80
Negative ind. mean pressure, p_i- lbs. per sq.in.	5.13	5.03	6.74
Net ind. mean pressure, $p_i=p_{i+}-p_{i-}$ lbs. per sq.in.	61.97	62.47	21.06
I.H.P. (from p_i) $=N_i$	79.50	80.00	27.00
Friction horse-power, N_i-N_e	10.00	10.55	—
Mechanical efficiency, %	87.5	86.9	—
Gas consumption per B.H.P. hour, cu.ft.	78.4	78.6	—
Gas consumption per I.H.P. hour, cu.ft.	68.6	68.3	107.5
Heat value of gas (32° and 760 mm.) B.T.U. per cu.ft	112.0	113.0	101.0
Heat used in gas, per B.H.P. hour, B.T.U.	8 770	8 850	—
Heat used in gas, per I.H.P. hour, B.T.U.	7 650	7 680	10850
Consumption of brown coal per B.H.P. hour, lbs.	1.865	1.61	—
Consumption of brown coal per I.H.P. hour, lbs	1.63	1.40	—
Heat used in coal per B.H.P. hour, B.T.U.	14 250	12 320	—
Heat used in coal per I.H.P. hour, B.T.U.	12 570	10 680	—
Volume of gas per lb. of coal, cu.ft.	42.1	48.5	43.2
Efficiency of producer, %	61.3	71.6	56.6
Total economic efficiency of installation, %	17.75	20.5	—

The brown coal used, Agnesschacht mine, had an average heating value of 7670 B.T.U./lb. The composition of the gas made is shown in the following table:

Test No.	I.	II.	III.
CO_2, % by vol.	17.76	14.71	14.31
O, "	.24	.41	1.12
H_2 "	17.76	17.57	14.08
CH_4 "	2.03	1.36	1.71
CO "	9.99	12.14	9.33
N "	54.90	54.35	59.45
	100.00	100.00	100.00
Heating value by calculation, B.T.U. per cu.ft.	112.0	113.0	101.1

The recent catalogues of the Deutz Co. contain the following additional tests on brown coal suction-gas plants:

Rated B.H.P.	60		160	
Revolutions per minute	144.1	180.67	—	—
B.H.P.	65.4	98.5	167.7	101.1
Coal consumption per B.H.P. hour, lbs.	1.218	1.115	1.277	1.338
Heating value of brown coal briquettes, B.T.U. per lb.	9200		9180	

(*f*) *Petroleum and Gasoline Engine*, 6 B.H.P., $D=6.7''$, $S=10.2''$, horizontal type. Compression pressure appr. 57 lbs. Hit-and-miss speed regulation, and electric ignition. Tests made by Prof. E. Meyer, 1899, alternately with gasoline and kerosene, showed results as per table following:[1]

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[1] Z. d. V. D. I., 1900, p. 330.

TABLE 47

Fuel.	Gasoline.			Kerosene.		
Load.	Full.	Half.	No Load.	Full.	Half.	No Load.
Revolutions per minute	242.0	244.9	247.3	240.2	244.2	246.7
Ignitions per minute	110.2	56.6	19.5	109.5	58.4	12.4
B.H.P.	6.93	3.02	0	6.87	3.02	0
Fuel consumption per B.H.P. hr., lbs.	.681	.950	1.53 per hr.	.923	1.271	1.48 per hr.

(*g*) *Alcohol Engine* (Locomobile), 14 B.H.P., $D=8.28''$, $S=11.80''$, ratio of compression $\varepsilon=5.91$. Speed regulation by changing quantity of charge (taper cam for inlet-valve lever), electric ignition. Internal vaporization, that is without special vaporizer, starting with gasoline.

Various tests made by Prof. E. Meyer, in March and May, 1901, showed the results given in the following tables. For the first the fuel was 87.2 weight per cent alcohol, having a lower heating value of 10440 B.T.U. per lb.

TABLE 48

Test No.	Load.	R.p.m. n	Compression, Lbs.	B.H.P.	Alcohol per B.H.P. Hour. Lbs.	Economic Efficiency. %	Outlet Temperature of Cooling Water.		
							Cylinder Head.	Cylinder Jacket.	Average of Both.
1	max.	275.2	107	15.68	.960	25.3	208.4	172.0	201.2
2	normal	280.3	100	13.91	1.00	24.3	208.4	179.5	208.4
3	$\frac{3}{4}$	279.1	—	10.42	1.060	23.1	208.4	170.5	203.0
4	$\frac{1}{2}$	282.7	53.2	7.05	1.218	20.0	208.4	176.0	208.4
5	$\frac{1}{4}$	291.4	—	3.64	1.885	13.0	208.4	172.0	208.4
6	no load	293.5	21–15	—	5.85 per hr.	—	208.4	201.2	208.4

The consumption of cooling water varied from 21.6 to 19.5 lbs. per B.H.P. hour, being small because of the high outlet temperature.

Tests with alcohol-benzol mixtures made by the same experimenter on the same engine gave the following results:

TABLE 49

Kind of Fuel.	Commercial Alcohol 86.4 Wt. % Alcohol and 13.6 Wt. % Water. $H_u=10\,350$ B.T.U.			Mixture: 9.12 Wt. % Benzol and 90.88 Wt. % Alcohol. $H_u=10\,980$ B.T.U.			Mixture: 14.3 Wt. % Benzol and 85.7 Wt. % Alcohol. $H_u=11\,340$ B.T.U.			Mixture: 20.95 Wt. % Benzol and 79.05 Wt. % Alcohol. $H_u=11\,808$ B.T.U.					
Test No.	2	1	3	7	8	9	4	5	6	10	11	12	13	14	15
Load.	Maximum.	Normal.	$\frac{1}{2}$	Maximum.	Normal.	$\frac{1}{2}$	Maximum.	Normal.	$\frac{1}{2}$	Maximum.	Normal.	$\frac{3}{4}$	$\frac{1}{2}$	$\frac{1}{4}$	No Load.
R.p.m.	280	273.3	286	277.5	278	286.7	275.8	278.3	287.2	279.1	280.2	286	287.8	291.7	297.3
B.H.P.	15.92	13.60	7.20	15.78	13.87	7.21	15.68	13.90	7.24	15.85	14.00	10.72	7.24	3.72	—
Fuel per B.H.P. hr., lbs.	.993	1.033	1.175	.942	.968	1.16	.865	.918	1.12	.837	.859	.887	1.17	1.47	— 3.82 per hr.
Econ. eff.,%	24.7	23.8	20.8	24.5	23.8	19.8	25.85	24.3	20.0	25.7	25.0	24.3	20.1	14.7	—

(*h*) In the main trials of alcohol locomobiles, made under the auspices of the *Deutsche Landwirtschaftliche Gesellschaft* in 1902, a Deutz engine gave the results in the following table, compiled from the report of Prof. Eugen Meyer. The machine was of the horizontal type, $D=8.30''$, $S=11.8''$, ratio of compression $\varepsilon=8.90$, piston speed 9.2 ft. per sec. The engine had no special vaporizer, and was regulated by throttling the charge.

TABLE 50

Load.	R.p.m.	B.H.P.	Consumption per B.H.P. Hour.			Economic Efficiency, %	Volume of Charge in % of Cylinder Volume.	Maximum Suction Pressure, lbs. per sq.in.	Compression Pressure, lbs. per sq.in.	Explosion Pressure, lbs. per sq.in.	Exhaust Temperature, Deg. F.
			86.1 Wt. % Alcohol. lbs.	Cylinder Oil. lbs.	Jacket Water, lbs.						
Maximum . .	276.9	16.55	.813	.0057	1.69	31.6	93	− 2.40	193.0	468.0	>900
Normal	284.1	11.93	.867	—	2.10	29.6	71	− 5.23	136.0	270.0	892
Half	292.5	6.18	1.115	—	2.60	22.7	42	− 9.08	76.5	178.0	892–897
No load	298.2	—	4.63 per hr.	.0136	—	—	19	−11.50	36.8	113.5	585–625

The alcohol used was 86.1 weight-per-cent, for which Meyer computed a heating value of $(.861\times 11\,664)-(.139\times 1080)=10\,043-150=9893$ B.T.U. per lb.

2. Gebr. Körting A–G in Hannover (See also Plates XII to XV).

This firm makes the *single-acting 4-cycle* machine shown in Fig. 486, in sizes from 2 to 175 H.P.; from 175 to 350 H.P. the same design is used as a twin engine. The double-acting 2-cycle engine shown in Fig. 493, p. 338, is made in sizes from 350 H.P. up.

The most important details of the 4-cycle design are set forth in Figs. 487 to 492 following. With reference to Figs. 490–492, *a* is the inlet valve operated by cams and levers *a′* and *b′* respectively. Mixing valve *c* is automatic, while the starting valve *d* is operated by hand by the lever *d′*. Electric ignition *e* is double. Combustion space and cylinder are connected only by the comparatively narrow passage *f*, formed by the vertical projection *g*, which is effectively water-cooled. The idea of this construction is to increase the cooling surface for the purpose of raising the compression. At the lowest point of the cylinder there is provided a blowing-off and drain cock *h*. The speed of these engines, as in all machines made by this firm, is controlled by throttling the charge—in this case by a butterfly throttle-valve placed in the inlet passage between the mixing valve and the inlet valve.

In the later construction, Figs. 487–489, the vertical projection *g* has been replaced by a horizontal water-cooled partition, again for the purpose of increasing the cooling surface and decreasing the end temperature of compression. This partition forms a part of the cover to the cylinder head proper, and this construction makes it comparatively easy to get at the interior of the cylinder without removing the piston. Above 70 H.P. the pistons are lined with white metal, and above 100 H.P. the pistons are water-cooled. All of the sizes, with the exception of the small models up to 8 H.P., have an outboard bearing. Below 12 H.P. the electric ignition is replaced by the open hot tube.

The *double-acting 2-cycle* engine, Fig. 493, as stated, is not made below 350 H.P. Its method of operation is in general as follows (see Figs. 494 and 495):

The main cylinder *a* contains a water-cooled piston *b* the length of which is about $\frac{7}{8}$ that of the stroke. A side crank, 110° ahead of the main crank, by means of common gearing operates an air pump *c* and a gas pump *d*, which are controlled by piston valves *c′* and *d′* respectively. From each pump separate ports lead to the inlet valves *e* and *e′*, in the housing of which the mixture is formed. The ring of ports at the middle of the cylinder serves as the exhaust outlet for both ends of the cylinder, alternately on one and then on the other side of the piston *b*. At the moment that the piston commences to cover the exhaust ports, the pump pistons have already completed about half of their stroke. Up to this instant, the suction passage of the gas pump *d* has been kept open, thus allowing about 40% (at full load) of the gas to flow back into the suction mains. After this the valve *d′* closes the suction passage and opens the discharge passage, and the main cylinder *a* commences to take its charge. In the mean time the air pump *c* has delivered air into its discharge passages, or into a receiver in the base of the machine, from the beginning of its stroke and has forced some of this air up into the gas passage next to the inlet valve housing *e*. Therefore, when the inlet valve *e* is opened, air will first flow out of both passages into the cylinder to be soon afterward followed by the mixture formed in the housing. This scheme gives a preliminary charge consisting of air only, which is desirable on several accounts. Scavenging and charging continue until the pump pistons have reached their dead centers, which happens at the instant the piston *b* has covered the exhaust ports *f*. The mixture is next compressed and then ignited in two places simultaneously.

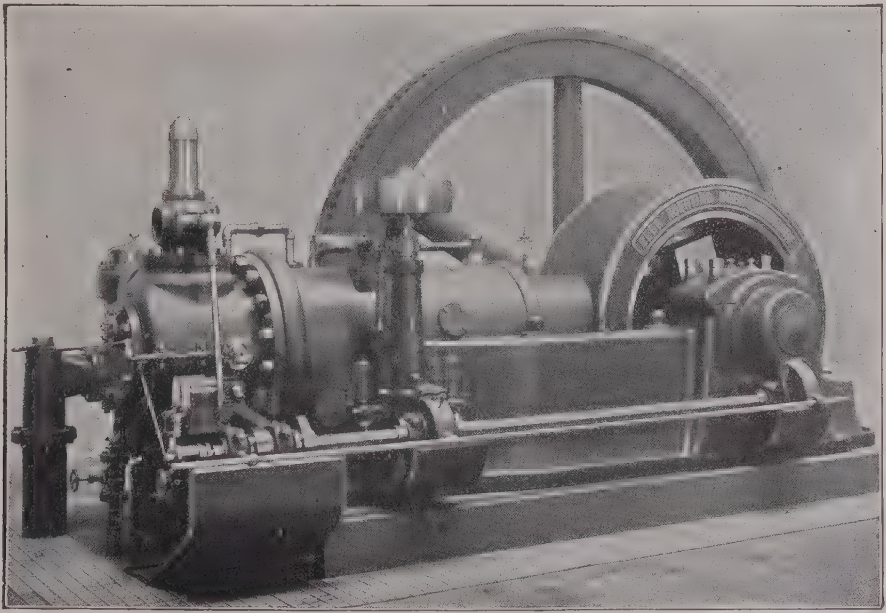
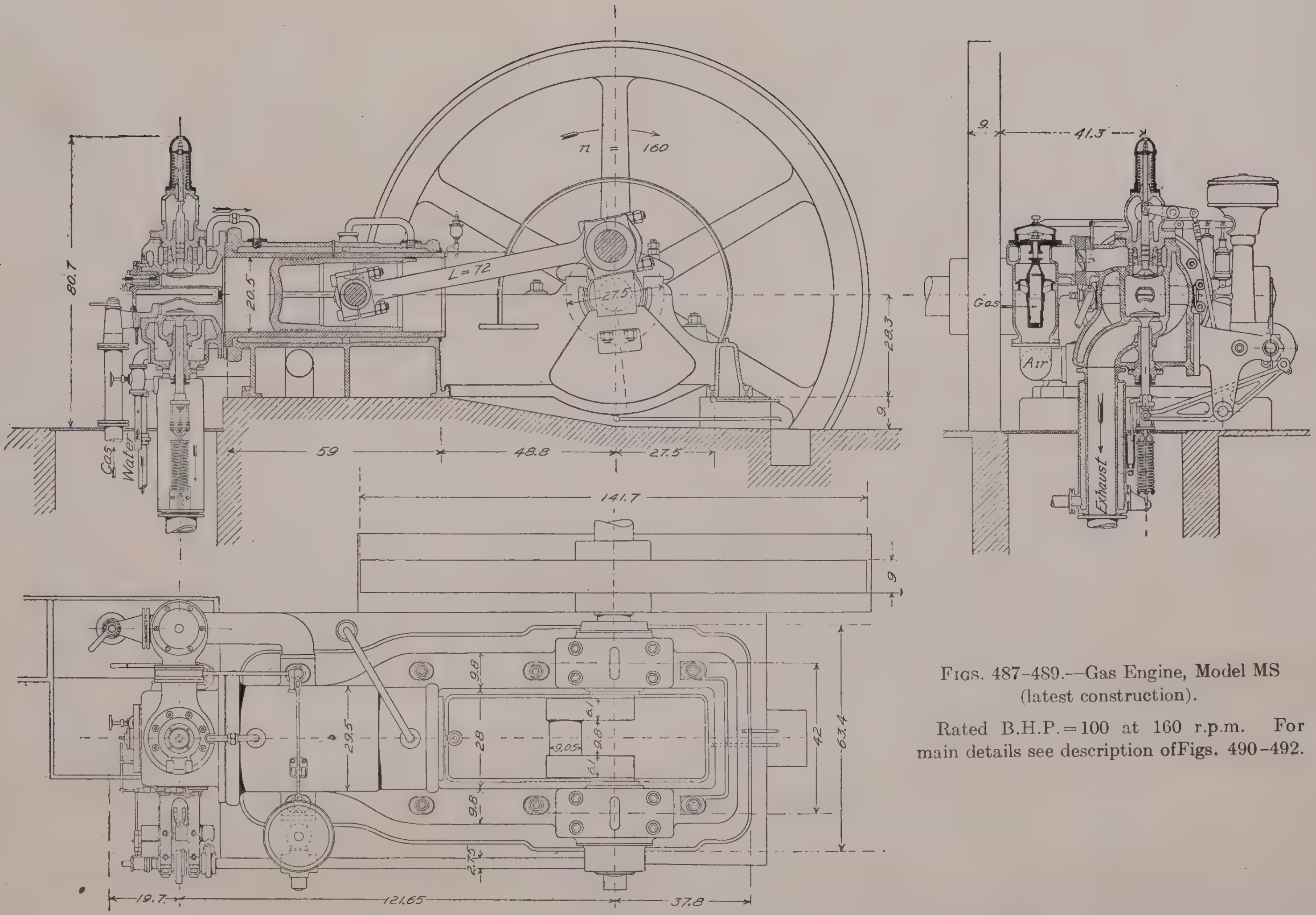


FIG. 486.

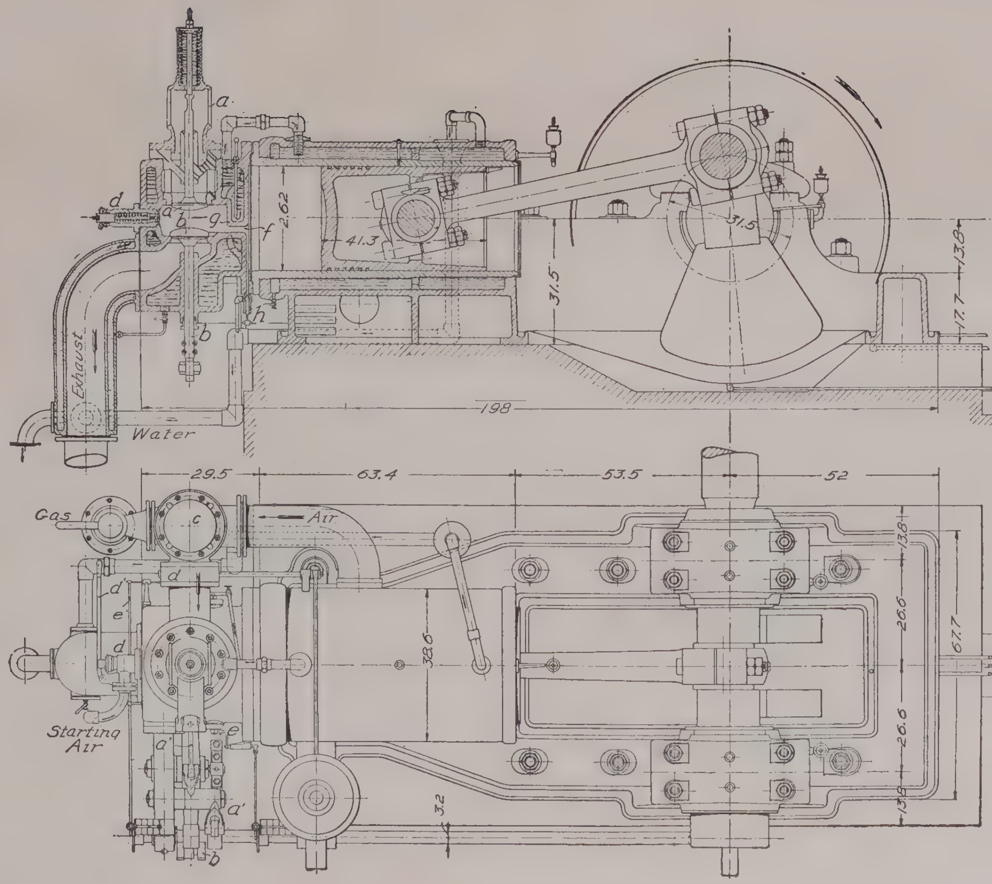
The principal features of the construction are shown in Plates XII to XV.

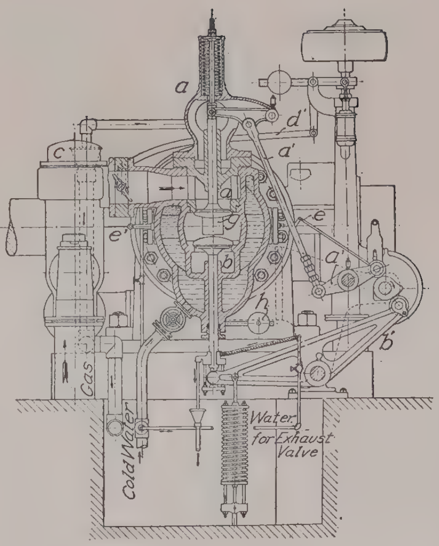
Speed regulation is effected by proportioning the quantity of gas to the load by means of changing the effective delivery stroke of the gas pump. For this purpose the hollow gas pump piston valve *a*, Figs. 496 and 497, contains a two-part piston valve *d* which controls the beginning of the delivery period in the stroke of the pump. The main valves *a* and *b* of the gas and air pumps respectively are carried by the common hollow valve rod *c*, which by means of the rocker arm *c′* is connected to the first eccentric. The auxiliary valve *d* of the gas pump is connected to the second eccentric by means of *e* and *e′*.



FIGS. 487–489.—Gas Engine, Model MS (latest construction).

Rated B.H.P. = 100 at 160 r.p.m. For main details see description of Figs. 490–492.





Figs. 490–492.—Gas Engine, Model MS. Rated B.H.P. = 160 at 140 r.p.m.

Projection *g* is no longer used, but has been replaced by an axial intermediate wall, see Fig. 487 on preceding page.

The air-pump valve controls the suction and delivery of air independent of the load, making the alternate connections near the end of the pump stroke in each case. Thus the same quantity of air is handled at all loads. Gas valve *a*, on the other hand, keeps the suction passages of the gas pump open far into the stroke, so that of each charge at least one-half is forced back into the suction mains. At maximum load, valves *a* and *d* close the suction passages at about half stroke, opening the delivery passages immediately thereafter. At this instant the main crank, 110° behind the pump crank, passes through its dead center. With decreasing loads, the governor delays the action of the auxiliary valve *d*, allowing the pump to force more gas back into the mains and less into the cylinder. A modification of this scheme consists in allowing the pump to draw a part of the charge out of the pressure passages leading to the cylinder, thus controlling the quantity of fresh gas. The pressure in the pumps at the end of the delivery stroke goes up to 8.5 lbs. gauge.

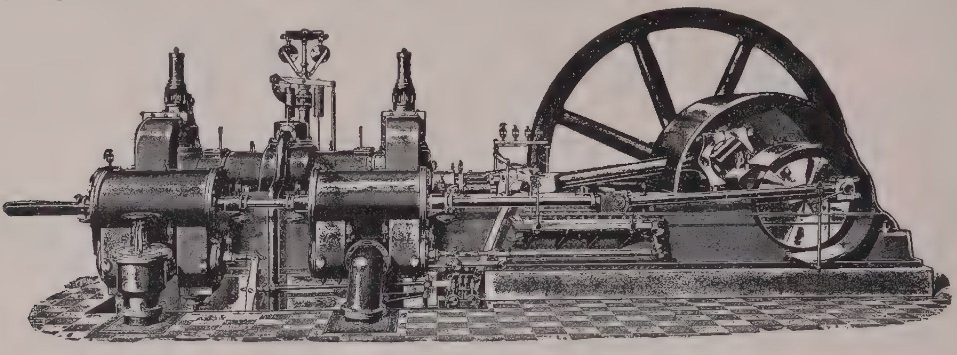
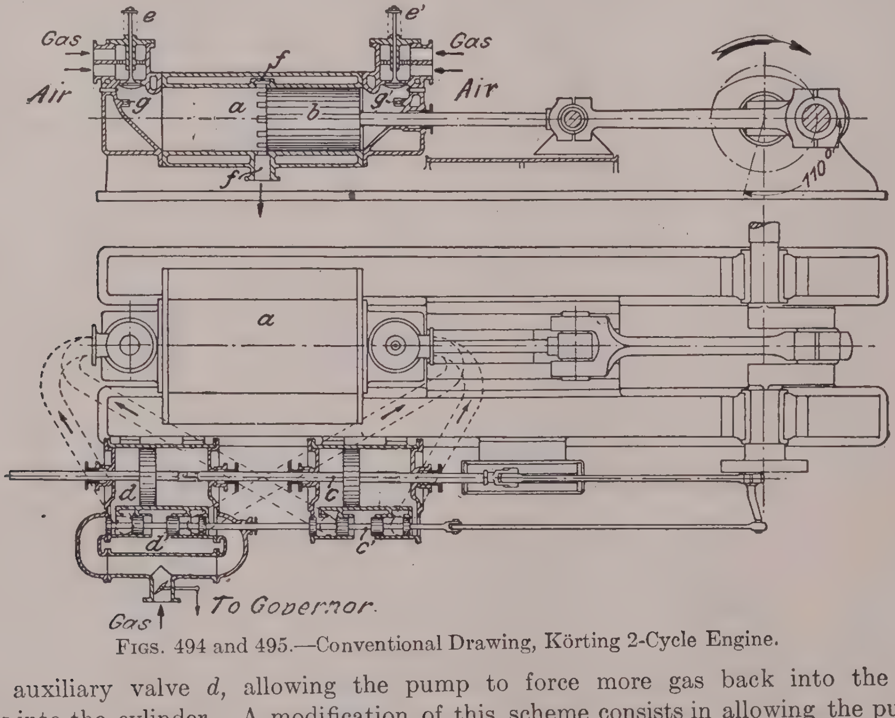
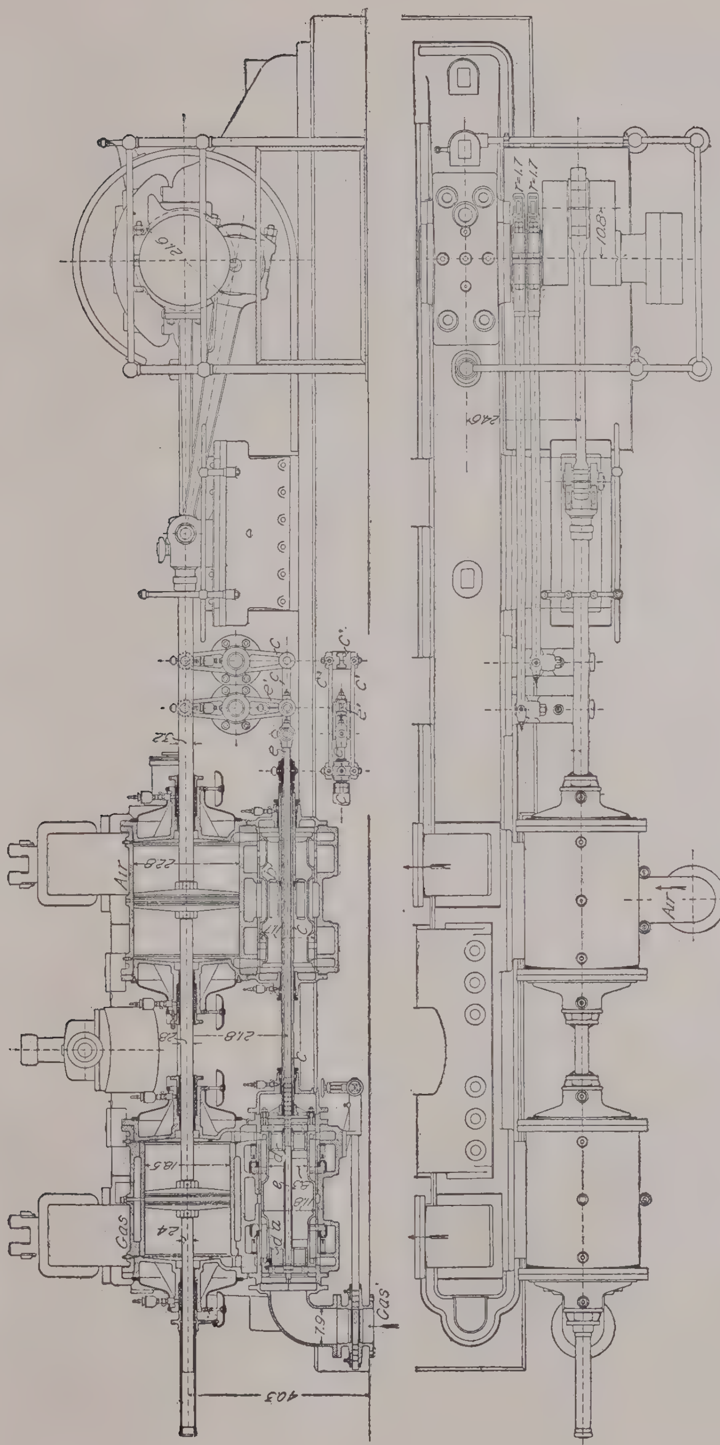


Fig. 493.



Figs. 494 and 495.—Conventional Drawing, Körting 2-Cycle Engine.

The air forced up into the gas passages next the inlet valve, as above described, is intended first to act as a scavenging agent, and secondly to aid in "stratifying"



Figs. 496 and 497.—Charging Pumps of a Körting 2-Cycle Engine. (Pump Crank leads Main Crank about 110°.)

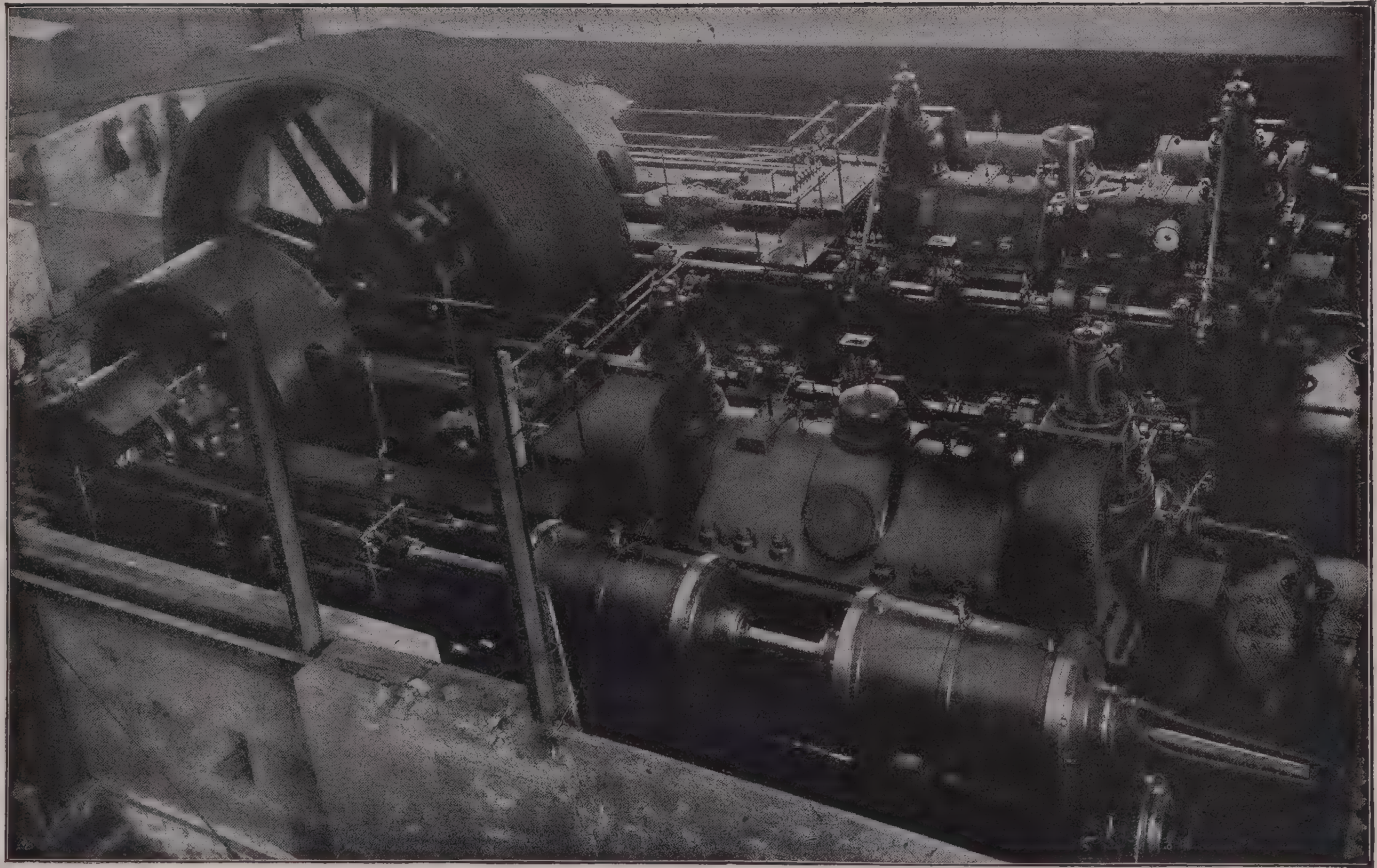
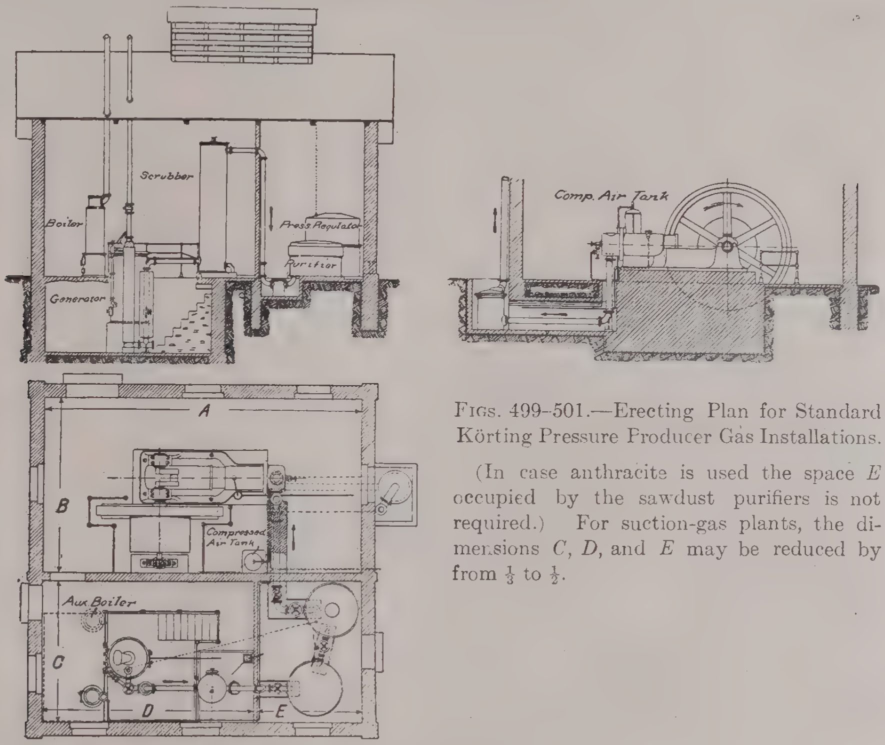


FIG. 498.—Two-Cycle Engine, Körting Type. Rated B.H.P. = 1800; Cyl. Diameter 30.5″; Stroke 51.2″; R.p.m. = 100. Built by the Maschinenbau-Aktiengesellschaft vorm. Gebr. Klein, Dahlbruch.

the charge in the cylinder. There is no doubt that by this means a comparatively lean mixture may be obtained next to the piston.*



Figs. 499–501.—Erecting Plan for Standard Körting Pressure Producer Gas Installations.

(In case anthracite is used the space *E* occupied by the sawdust purifiers is not required.) For suction-gas plants, the dimensions *C*, *D*, and *E* may be reduced by from $\frac{1}{3}$ to $\frac{1}{2}$.

Table 51

DIMENSIONS OF KÖRTING PRESSURE GAS INSTALLATIONS

N_n (in B.H.P.)		12	14	16	20	25	30	35	40	50	60	80
Number of producers		1	1	1	1	1	1	1	2	2	3	3
		Single-unit Plants										
Dimensions in feet	*A*	15.6	16.4	17.0	17.7	18.4	20.3	21.6	22.0	23.0	24.2	26.2
	B	11.2	11.5	11.5	12.5	12.5	12.8	13.1	14.1	14.7	15.7	16.4
	C	10.5	10.5	10.5	10.5	10.5	10.5	10.5	11.8	11.8	13.1	13.1
	D	17.4	17.4	17.4	17.4	17.4	17.4	17.4	18.3	18.3	19.7	19.7
	E	7.9	7.9	7.9	7.9	7.9	7.9	7.9	8.8	8.8	9.8	9.8
		Two-unit Plants										
Dimensions in feet	*A*	15.6	16.4	17.0	17.7	18.4	20.3	21.6	22.0	23.0	24.2	26.2
	B	22.4	23.0	23.0	25.0	25.0	25.6	26.2	28.2	29.4	31.4	32.8
	C	14.8	14.8	14.8	14.8	14.8	14.8	14.8	16.4	16.4	18.0	18.0
	D	13.1	13.1	13.1	13.1	13.1	13.1	13.1	15.7	15.7	18.0	18.0
	E	9.8	9.8	9.8	9.8	9.8	9.8	9.8	10.5	10.5	11.8	11.8

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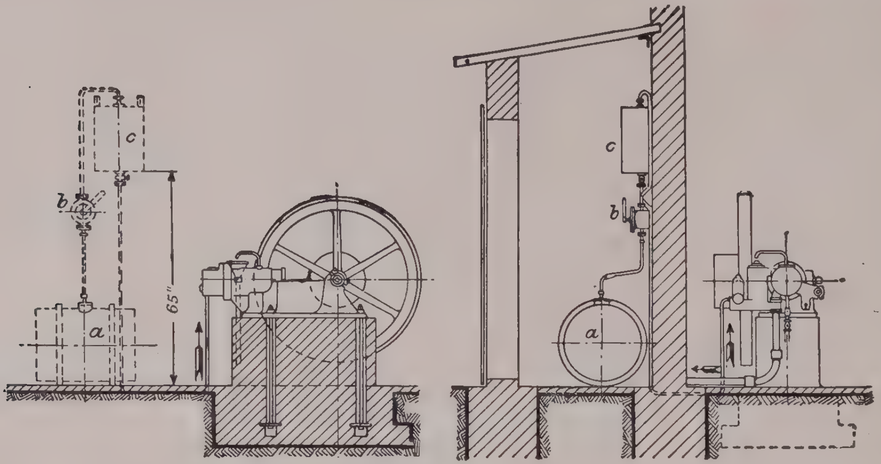
* See the extended discussion by Reinhardt concerning charging and governing phenomena in Stahl und Eisen, 1902, No 21.



FIGS. 502 and 503.—Erecting Plan for a Körting Gasoline Engine, Model M–A, with Spraying Carburetor. (Room for the storage tank a and the feed tank c is separated from the engine room by a heavy wall. Tank c is filled by means of pump b.)

Operating Results. (*a*) *Illuminating Gas Engine*, 8 B.H.P., $D=6.90''$, $S=13.4''$, compression ratio $\varepsilon=6.41$. Magneto Ignition, speed regulation by varying charge volume. Tests by Prof. E. Meyer.[1]

The illuminating gas used had a heating value of 496 B.T.U. per cu.ft., at 66.2° F. and 751 mm. bar., the consumption figures of the table, however, have been recomputed to the basis of gas of 562 B.T.U./cu.ft., at 32° F. and 760 mm.

TABLE 52

Test No.	Load.	R.p.m. n	B.H.P. N_e	Gas per B.H.P. Hour, cu.ft.	Compression Pressure, lbs. per sq.in.	Pressure at End of Suction, lbs. per sq.in.	Ratio Charge to Cylinder Volume, %.	Ratio Air to Gas.	Economic Efficiency, % η_w	Heat in Cooling Water, %.
1	full	221.2	10.27	16.05	139	13.8abs.	76	6.76	28	34.9
2	½	226.2	5.25	20.60	85	9.80	50	6.99	22	38.6
3	no load	229.3	0	57.2 per hour	35.5	6.35	23	6.00	—	—

The diagram obtained at maximum load showed an average mean effective pressure of 85 lbs. per sq.in. From this Meyer subtracts[2] the mean pressure of the lower loop diagram $p_{i-}=2.3$ lbs. per sq.in., and from the net pressure $p_i=p_{i+}-p_{i-}=85-2.3=82.7$ lbs. per sq.in., he computes the corresponding maximum I.H.P.$=11.55$

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[1] Z. d. V. D. I., 1900, p. 332.

[2] The correctness of this method of computation is just at present the subject of an extended controversy, published in the Zeitschrift des Vereins Deutscher Ingenieure (1905, pp. 324, 331, and 517) and started originally by an article due to Professor Riedler. At the present writing the conflicting opinions appear to hold a balance. The question is of importance in matters relating to the comparative performances of 2- and 4-cycle engines. It is quite likely that the issue will be settled by the code for the testing of gas engines now in preparation.

and the gas consumption per I.H.P.-hour = 14.4 cu.ft. Figs. 504 and 505 show two no-load diagrams obtained on Test No. 3.

The author has taken the following additional data on illuminating gas engines from the catalogues and other papers of the firm. The tests were all made by independent experts, and the figures for gas consumption have been referred to gas under standard conditions.

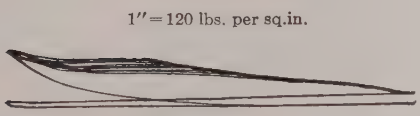

FIG. 504.

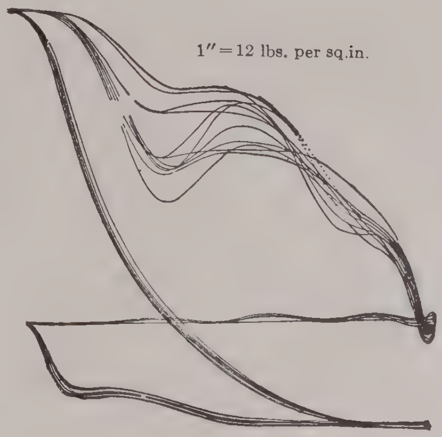

FIG. 505.

TABLE 53

100 B.H.P. Illuminating Gas Engine.	Electric Station at Göttingen.				Electric Station, Meissen.	
	No. 1.		No. 2.			
Load	full	$\frac{2}{3}$	full	$\frac{2}{3}$	full	full
B.H.P.	114.1	76.8	115.0	76.9	97.20	97.30
Gas per B.H.P. hour, cubic feet	14.10	16.38	15.32	17.75	14.70	14.05
Economic efficiency, %	32	27.5	29.5	25.0	30.5	32.3

(*b*) Two 125 B.H.P. pressure producer gas engines of the older type in the Electric Station at Erlangen gave the following results:

Engine No.	I	II
Mean load	83.93 K.W. = 125 B.H.P.	88.7 K.W. = 136 B.H.P.
Fuel consumed per K.W.-hr., boiler and producer, lbs.	1.230	1.29
or		
Gas coke and anthracite, per B.H.P. hour, lbs.	.837	.878
Cooling water per B.H.P. hour, lbs	73.6	72.5
Mechanical efficiency (n = appr. 126 r.p.m.), %	85	85

From the above figures we may obtain the following additional data:

I.H.P.	149	162.5
Fuel per I.H.P. hour, lbs.	.712	.746
Indicated thermal efficiency % (taking H_u = 13,500 B.T.U./lb.)	26.5	25.3
Economic efficiency of entire plant, %	22.5	21.4

The figures guaranteed by the makers for the above installation were: fuel consumption, 1.71 lbs. per K.W.-hour; mechanical efficiency, 80%; and cooling water consumption not to exceed 110 lbs. per B.H.P.-hour. The difference in speed between full load and no load was 2% of the rated speed; for sudden changes of load amounting to 25%, the variation in the speed amounted to not quite 1.5%. During the two-day tests each unit was temporarily loaded with 94 K.W. = appr. 140 B.H.P.

(*c*) A suction gas plant of 100 B.H.P. rated capacity, operated with brown coal briquettes tested in the summer of 1904 by R. Pawlikowski, showed the results given in the following table:

Cylinder diameter, 21.2″, stroke, 34.4″, r.p.m., 150.

Trial	I	II	III
Duration, hours	5	3	3
R.p.m.	150–132	152–148	156–154
B.H.P.	118.5–113.5	108.5	53.8–53.6
Suction vacuum, inches water	1.97–2.36	1.57–1.93	.97–1.38
Temperature of cooling water, outlet, degrees F	104–113	95–100	91–95
Weight of one full charge for producer, lbs.	154–198	154–176	88
Time of gasification per charge, hour	1	1	1
Fuel consumption per B.H.P. hour, lbs.	1.477	1.492	2.18
Fuel consumption per B.H.P. hour, lbs., guaranteed	1.562	1.562	2.39

At one-quarter load the fuel consumption was guaranteed not to exceed 3.19 lbs. per B.H.P.-hr. During stand-by periods lasting for 12 hours (over night) the producer burned from 99 to 110 lbs. of coal, which amounts to about 8% of the daily consumption at full load.

Kind of fuel: Brown coal nut briquettes, average size 2″, heating value 8607 B.T.U./lb.

3. Vereinigte Maschinenfabrik Augsburg und Maschinenbau-Ges. Nürnberg A.-G., Werk Nürnberg. (See Plates XVI to XVIII.)

The new model of large gas engine made by this firm and recently introduced operates on the 4-cycle double-acting principle and uses eccentrics to operate the valves. The constructive details of these engines are clearly brought out by plates following, and by Figs. 508 and 509. The sizes at present built and their main dimensions are compiled in the following table.

TABLE 54

STANDARD CAPACITIES AND MAIN DIMENSIONS OF NEW TYPE DOUBLE-ACTING LARGE GAS ENGINES

Single-cylinder units	D. 6*a*	D. 7	D. 7*a*	D. 8	D. 9	D. 10	D. 11	D. 12	D. 13
Rated B.H.P.	160	220	245	315	400	480	550	690	845
R.p.m.	150	150	125	125	120	110	110	94	90
Max. length to center of crank-shaft ft.	18.0	19.2	20.3	22.0	23.8	25.6	28.5	30.2	32.8
Diameter of fly-wheel ft.	13.5	16.1	16.1	16.7	16.7	18.1	20.0	21.3	22.1
Distance floor to center of shaft ft.	2.46	2.51	2.54	2.62	2.79	2.95	3.12	3.20	3.28
Maximum width* ft.	14.7	15.4	18.1	19.7	20.5	21.4	22.1	23.0	23.8
Tandem units	D. T. 6*a*	D. T. 7	D. T. 7*a*	D. T. 8	D. T. 9	D. T. 10	D. T. 11	D. T. 12	D. T. 13
Rated B.H.P.	350	480	530	685	870	1050	1200	1500	1850
R.p.m.	150	150	125	125	120	110	100	94	90
Max. length to center of crank-shaft ft.	28.0	30.3	32.8	35.2	38.2	41.0	44.3	46.7	49.2
Diameter of fly-wheel ft.	13.5	16.1	16.1	16.1	16.7	18.1	20.0	21.3	22.1
Distance floor to center of shaft ft.	2.46	2.51	2.54	2.62	2.79	2.95	3.12	3.20	3.28
Maximum width* ft.	18.0	18.7	19.7	20.5	21.3	22.1	23.0	23.8	24.6

* To outer edge of out-board bearing.

The sketches shown in Figs. 506 and 507 indicate the manner of opening up the interior of the cylinder and the taking out of the pistons in a Nürnberg tandem engine.

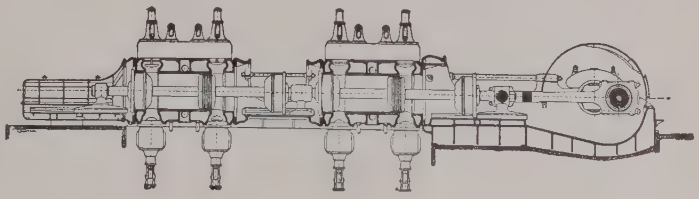

FIG. 506.—Method of Opening up the Front End Combustion Chambers.

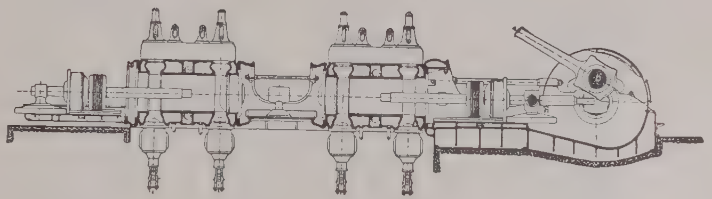

FIG. 507.—Taking the Pistons out of a Tandem Engine.

Operating Results. Recent information on fuel consumption and capacity obtained by independent observers is lacking in the data available. The heat consumption at full load for the large machines is given at from 8700 to 10 300 B.T U. per B.H.P.-hour. This, if a gas producer is used, would mean a consumption of from .815 to .945 lbs. of anthracite and from 1.00 to 1.17 lbs. of coke per B.H.P.-hour. The consumption figures for $\frac{3}{4}$ and $\frac{1}{2}$ loads are stated to be 10 and 30% higher respectively than at full load, which would be a very remarkable showing.[1] The diagram, Fig. 510, was taken from a blast furnace gas engine.

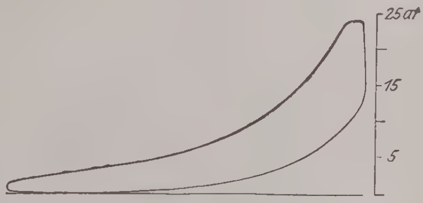

FIG. 510.

4. Deutsche Kraftgas-Gesellschaft m.b.H. in Berlin, (see also Plates XIX to XXII).

This company are the owners of the Oechelhauser 2-cycle engine patents. These engines are at present built in single cylinder sizes from 250 to 1000 B.H.P. and as twin units from 500 to 2000 B.H.P. The speed of all of the sizes is 125 r.p.m. The fuel is mostly blast-furnace gas. The 1000 B.H.P. twin cylinder engine for a coefficient of regulation of $\frac{1}{350}$ (operation of alternating current generators in parallel) has a fly-wheel weighing only 28 tons. The indicator card, when operating with blast-furnace gas, shows a compression pressure of about 150 lbs. and at maximum load an explosion

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[1] Since the printing of the above statement, Professor Riedler has published in the Zeitschrift d. V. D. I., 1905, p. 273, an article on "Large Gas Engines," from which the writer, among other things, takes the following results obtained in the operation of a double-acting tandem Nürnberg engine (No. IV, $D=33.5''$, $S=43.4''$, $n=106$):

$N_e=$	276	550	860	1024	1100	1131	1170 B.H.P.
$p_i=$	60.8	42.5	60.1	68.1	70.4	73.2	75.2 lbs. per sq.in.
$\eta_m=$	48.5	69.0	76.1	79.0	82.1	82.6	83.1%
Consumption =	18 800	12 400	10 920	10 120	9780	9320	9080 B.T.U. per B.H.P. per hr.

At the lowest load (276 B.H.P.) one cylinder was cut out. The power developed was measured electrically.

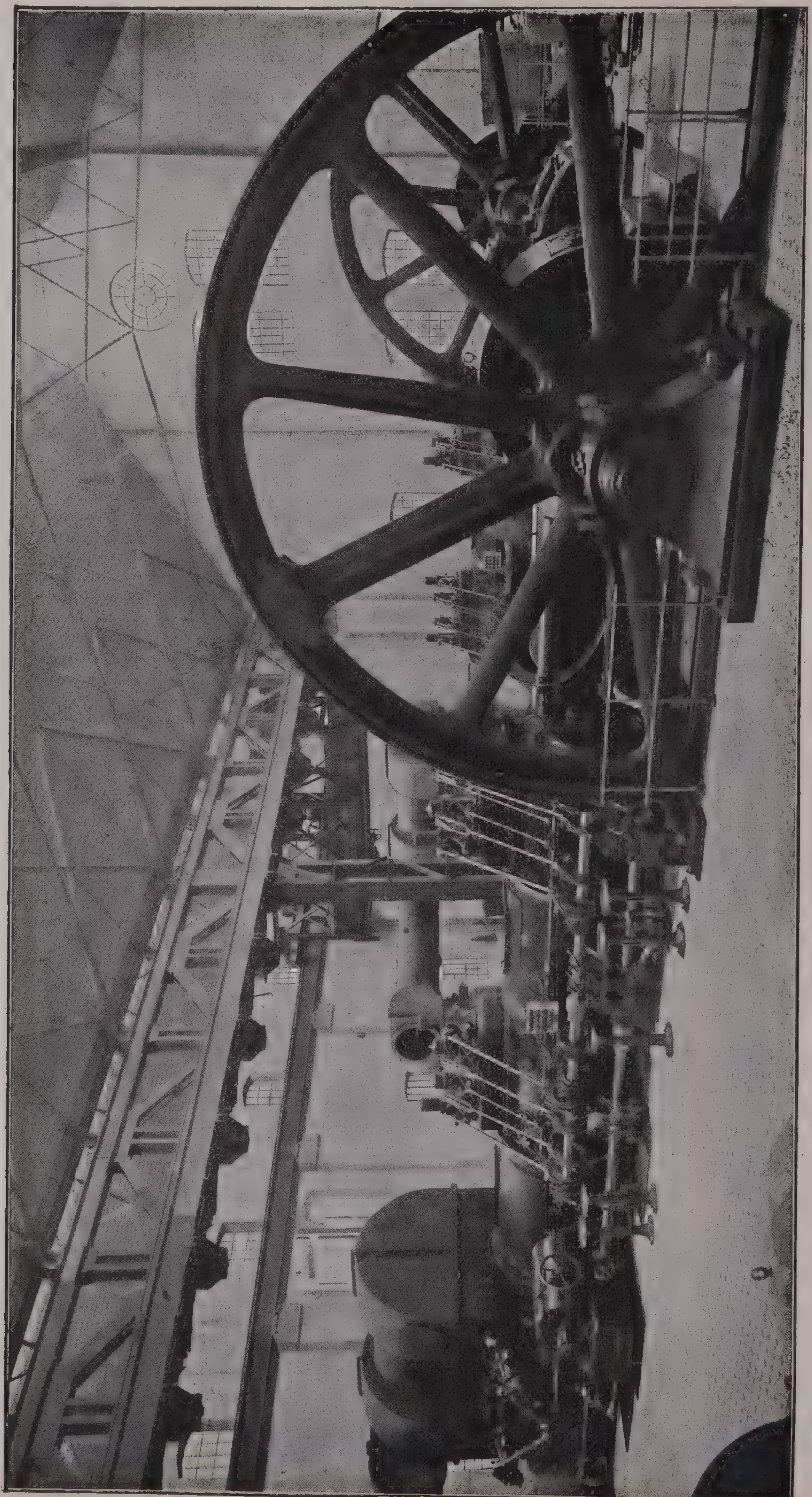


Fig. 508.—Nürnberg Tandem Engine direct connected to Blowing Cylinder. Effective H.P.=870 at 120 r.p.m. ($D=33.5''$, $S=43.4''$.) View of valve-gear side.

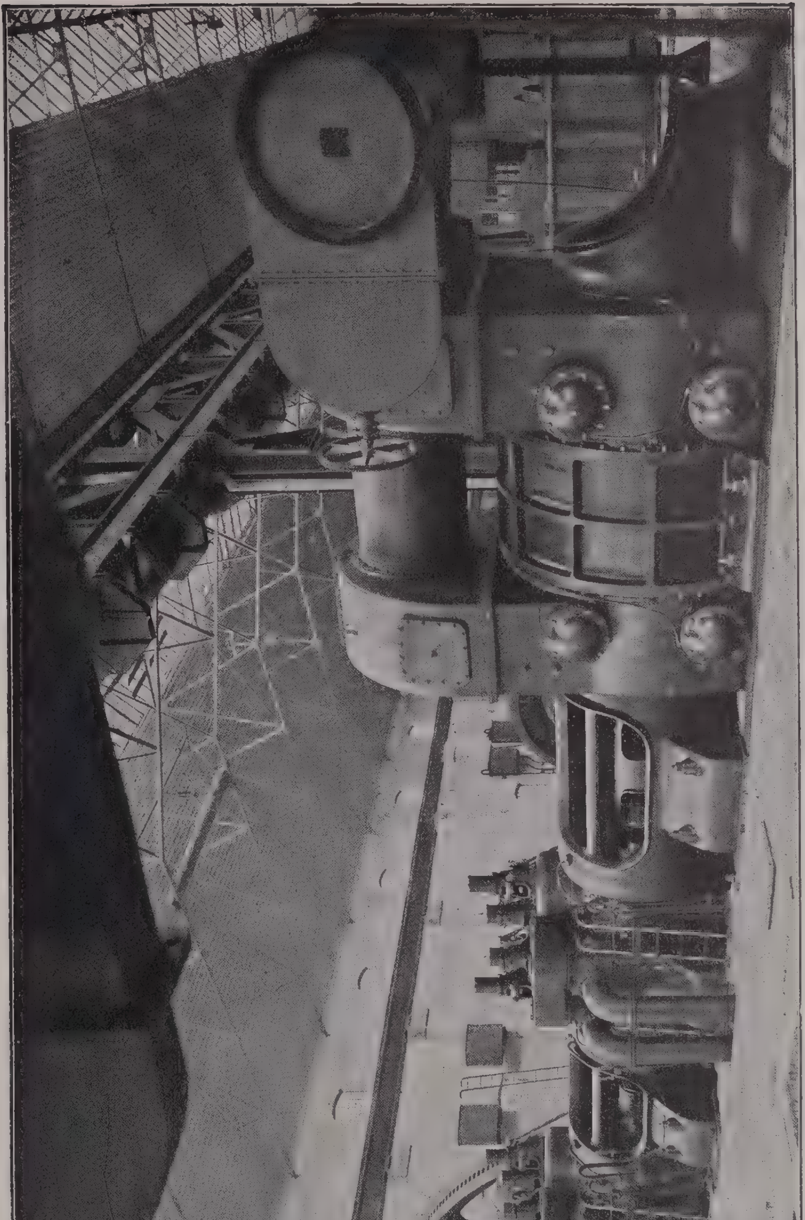
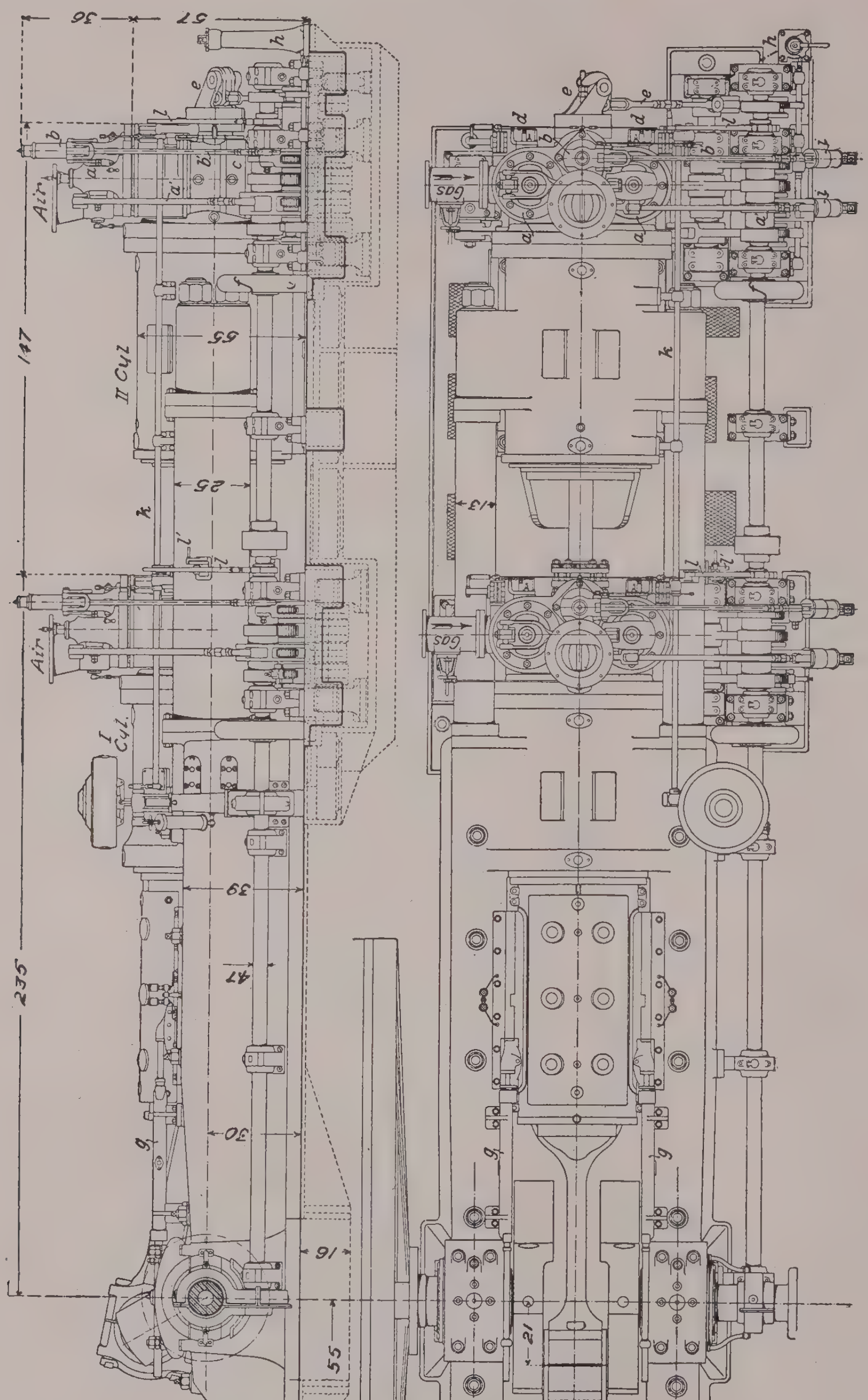


Fig. 509.—Nürnberg Tandem Engine, direct-connected to Blowing Cylinder. Effective H.P. = 870 at 120 r.p.m. (D = 33.5″, S = 43.4″.) View showing Frame Construction, Blowing Cylinder, etc.


36
57
147
235
Air
II Cyl
I Cyl
65
25
39
47
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16
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Gas
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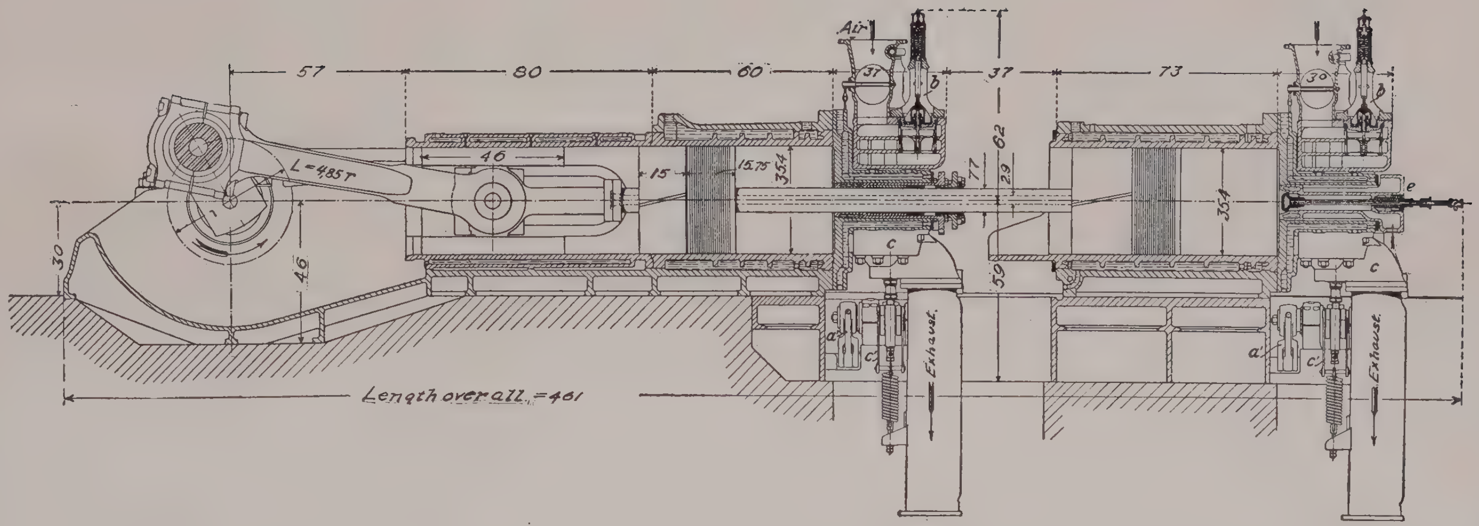
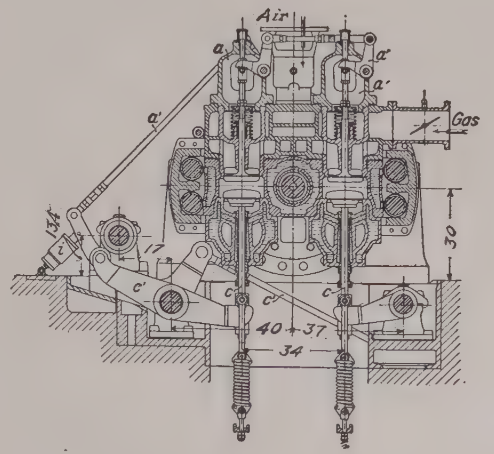
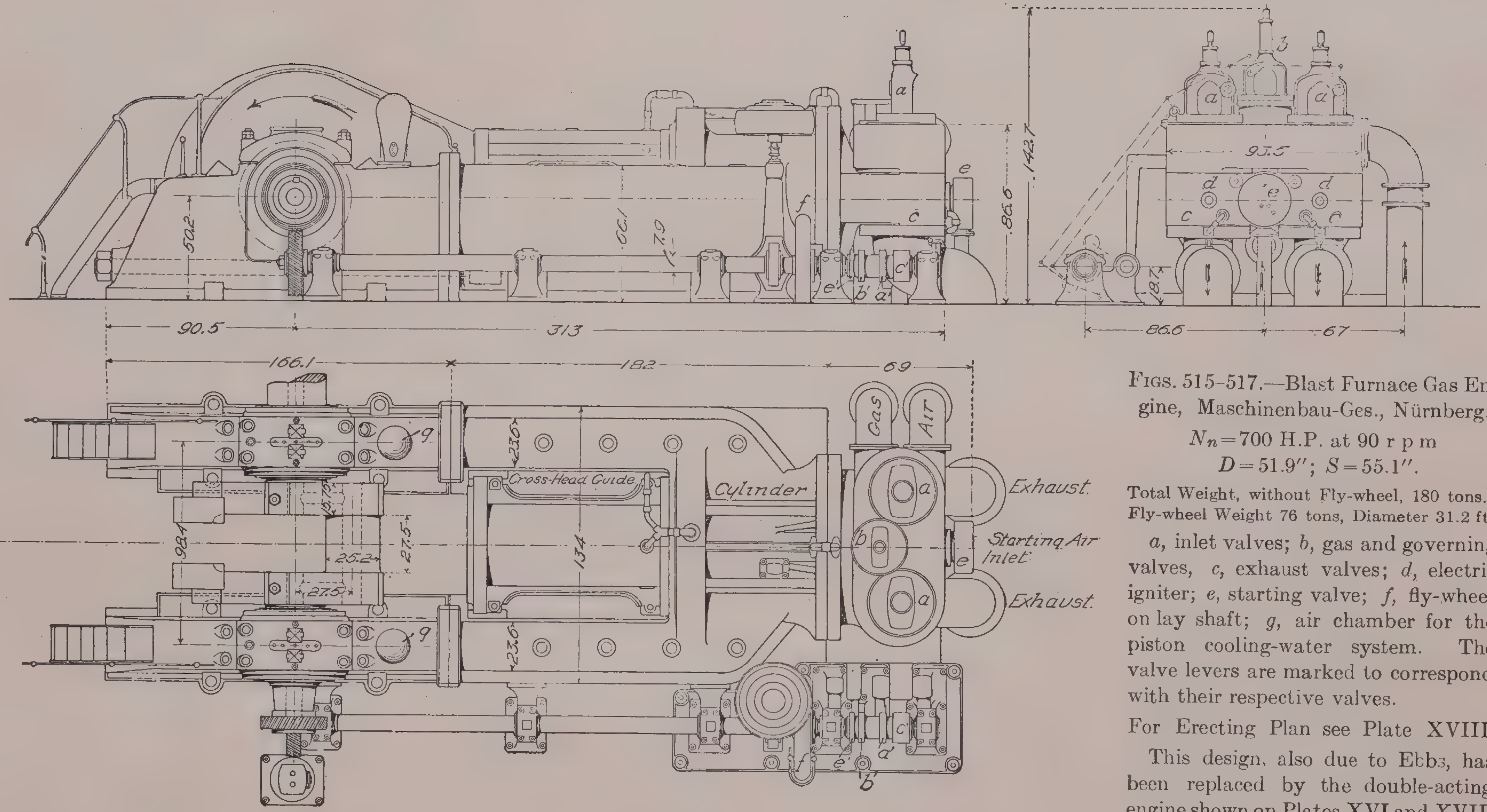




Figs. 511–514.—Tandem Producer Gas Engine, older Type, Maschinenbau-Ges. Nürnberg. $N_n = 800$ H.P. at 125 r.p.m.

a, inlet valves; *b*, mixing and governing valves; *c*, exhaust valves, *f*, lay-shaft fly-wheel; *g*, pipes with stuffing-boxes for supply and discharge of piston cooling water; *h*, support for starting gear; *i*, valve lever springs; *k*, governor linkage; *l*, igniter gear; *l'*, spark adjustment by hand. Speed regulation by changing the mixture (by means of the throttle valves) and by varying the quantity of charge (by means of sleeve valve *b*).

This design, due to Ebbs, has in the meantime been replaced by the double-acting engine illustrated on Plates XVI and XVII.



FIGS. 515–517.—Blast Furnace Gas Engine, Maschinenbau-Ges., Nürnberg.

$N_n = 700$ H.P. at 90 r p m
$D = 51.9''$; $S = 55.1''$.

Total Weight, without Fly-wheel, 180 tons.
Fly-wheel Weight 76 tons, Diameter 31.2 ft.

a, inlet valves; *b*, gas and governing valves, *c*, exhaust valves; *d*, electric igniter; *e*, starting valve; *f*, fly-wheel on lay shaft; *g*, air chamber for the piston cooling-water system. The valve levers are marked to correspond with their respective valves.

For Erecting Plan see Plate XVIII.

This design, also due to Ebbs, has been replaced by the double-acting engine shown on Plates XVI and XVII.

pressure of about 300 lbs. per sq.in. For method of speed regulation, see Plates XXI and XXII.

The method on which this engine operates is best explained from Plate XXII. The cylinder, a long straight tube open on both ends, contains two pistons. The front piston, by means of rod, cross-head, and connecting-rod, is connected to the middle crank of the three-throw shaft. The back piston, by means of a yoke, side rods, etc., is connected to the two outside cranks. This construction results in an engine almost perfectly balanced. The front piston controls the exhaust ports *a*, while the back piston uncovers first the scavenging air ports *b*, and a little later also the gas or mixture ports *c*. The air and gas pumps are in this case placed at the side of the engine and operated by a rocker arm and shaft, as shown. In some of the other constructions these pumps are operated by an extension of the back piston rod, or they may be placed under the floor and operated by rocker arm. The pumps supply gas and air under certain pressure to individual receivers which in turn connect with the gas and air ports *c* and *b* respectively. To explain the operation, suppose that both pistons have completed their instroke and have compressed between them the fresh charge (180° from the crank position shown in the plate). The charge is ignited electrically at *g* and the explosion forces the pistons apart. Near the end of the stroke, the front piston first uncovers ports *a*, and exhaust commences. A moment later the back piston uncovers ports *b*, admitting a charge of fresh air to sweep out the cylinder toward the front. This is next followed by gas from ports *c* to form the mixture. On the return the ports are covered in succession and the operation is repeated.

Operating Results. A 500-H.P. Oechelhäuser blowing engine in the Borsig Works was tested by Prof. E Meyer. The fuel was coke oven gas.

The machine was originally built by A. Borsig-Tegel for operation with blast-furnace gas. Hence to operate it with coke oven gas called for certain changes in pumps, ports, regulation, etc., which changes left certain imperfections in the construction, at times affecting the results unfavorably.

The main tests were made during the forepart of August. Certain check tests on the most important of the main trials were made at the end of October, 1904.*

MAIN DIMENSIONS OF ENGINE

Power cylinder	Diameter	26.6″	Air pump	Diameter	44.85″
	Stroke front piston	37.49″		Stroke	19.7″
	Stroke back piston	37.31″			
Blowing cylinder	Diameter	65.0″	Gas pump	Diameter	23.20″
	Stroke	37.31″		Stroke	19.7″

ANALYSIS OF THE COKE OVEN GAS USED (Vol. %)

CO_2	4.91	4.90	5.30
Heavy hydrocarbons	2.63	1.80	2.10
O_2	.20	.30	.40
CO	11.84	10.60	10.20
H_2	42.00	48.08	43.80
CH_4	19.73	18.43	20.30
N	18.69	15.89	17.90

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* The entire report has been published in Z. d. V. D. I., 1905, p. 324.

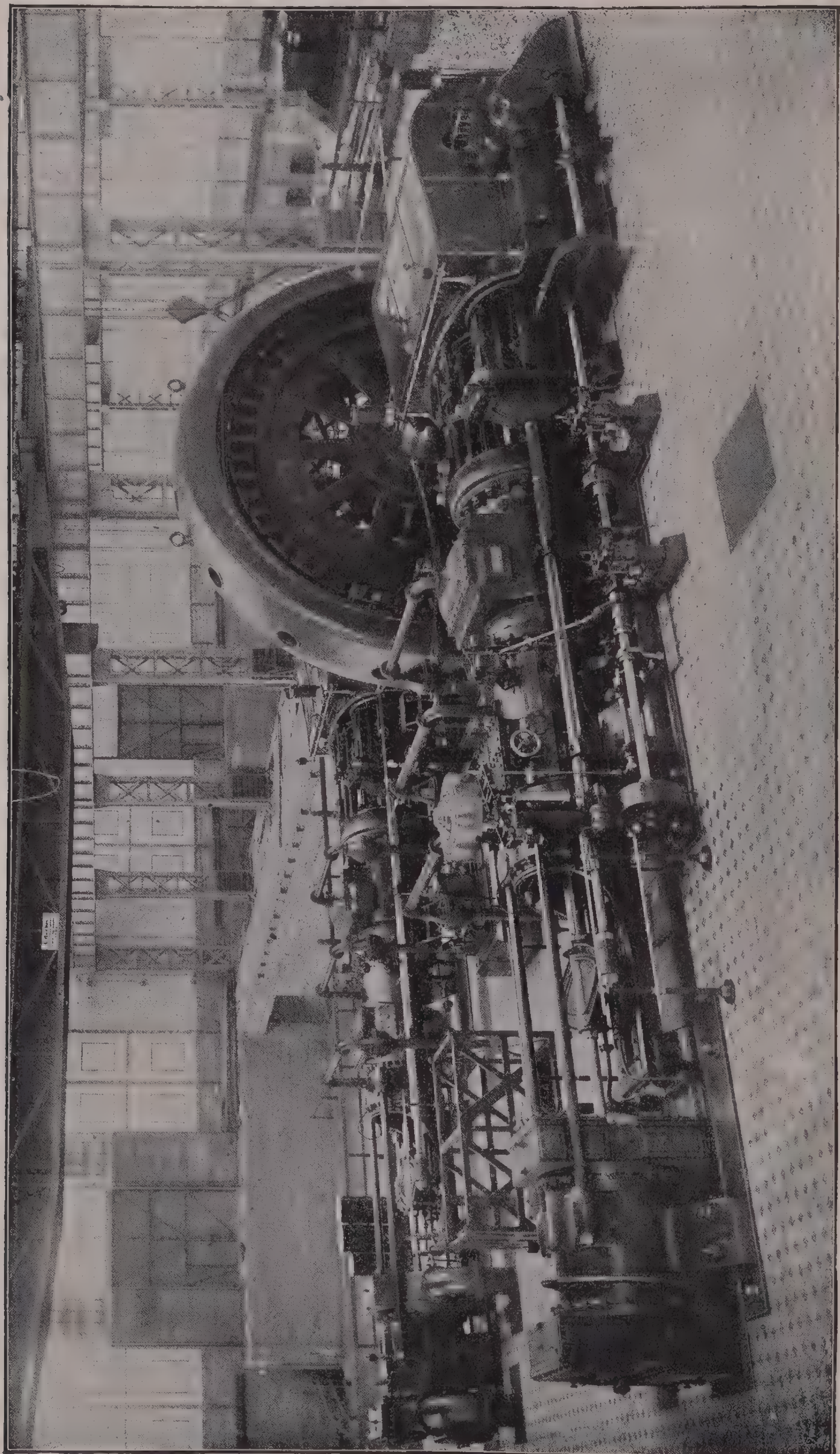


FIG. 518.—Twin Two-cycle Engine, Oechelhäuser Type. Rated B.H. P. = 1000 at 125 r.p.m., direct-connected to Alternating Current Generator. Built by the Aschersleben Maschinenbau-Akt.-Ges., Aschersleben.

The lower heating value of the gas was determined at short intervals of time by means of a calorimeter and was found to vary within the limits of 349 and 433 B.T.U. per cubic foot.

TABLE 55

PRINCIPAL DATA FROM SEVERAL OF THE TESTS

Tests Made	AUGUST 4, 5, 1904.								
Test Number.	VII*b*	XV	XIV	XIX	XVII	X	IV	VIII*b*	XII
R.p.m.	110.6	113.6	112.9	108.3	96.4	84.5	86.8	86.1	68.4
Compression, lbs. per sq.in.	153	150	150	143	145	143	139	139	142
Mean ind. pressure, lbs. per sq.in. p_{i+}	74.5	60.5	57.1	41.2	32.6	70.1	53.4	52.0	57.1
I.H.P. (from p_{i+}), N_{i+}	865	723	676	468	330	622	486	470	410
Mean ind. pressure in air pump, lbs. per sq.in. Front	7.3	6.7	6.8	5.06	5.85	6.93	4.70	4.60	4.10
Mean ind. pressure in air pump, lbs. per sq.in. Back	4.66	4.36	4.32	3.44	3.58	3.86	3.01	3.09	2.37
Air pump, I.H.P., N_l	104.0	98.6	98.7	72.5	71.5	71.7	52.6	52.1	34.8
Mean ind. pressure in gas pump, lbs. per sq.in.	3.11	3.45	3.52	3.56	3.69	3.53	3.15	3.34	3.28
Gas pump, I.H.P., N_g	7.25	8.25	8.37	8.14	7.51	6.29	5.76	6.06	4.73
Total pump H.P., $N+N_g$	111.2	106.8	107.0	80.6	79.0	78.0	58.3	58.6	39.5
Indicated engine H.P. $= N_i = N_{i+} - (N_l + N_g)$*	754	616	569	387	251	544	428	412	370
I.H.P. of blowing cylinder, N_w	619	476	433	269	144	457	343	338	293
Ratio $\frac{N_l + N_g,}{N_i}$ %	14.7	17.3	18.8	20.8	31.4	14.3	13.7	14.1	10.6
Total eff.: $\eta = \frac{N_w}{N_{i+}}$ %	70.3	65.7	64.1	57.5	43.6	73.5	70.6	71.8	71.3
Mech. eff.: $\eta m = \frac{N_w}{N_i}$ %	82.1	77.2	76.1	69.4	57.3	84.0	80.2	81.8	78.9
Work of friction, $N_r = N_i - N$	135	140	136	118	107	87	85	74	77
Heat used per I.H.P. hour:									
Based on N_{i+} B.T.U.	6700	6500	6500	7060	9630	7830	6980	6670	6820
Based on N_i B.T.U.	7710	7670	7740	8540	12610	8970	7950	7630	7540
Based on N_w B.T.U.	9400	9900	10140	12270	22100	10610	9900	9310	9580

* See foot-note, p. 342.

In the second series of tests, October, 1904, the best figure obtained for the heat consumption was 7310 B.T.U. per H.P. hour, based on N_i, and 8760 B.T.U., based on N_w. The corresponding thermal efficiencies 34.8 and 29% respectively, are extraordinarily good. The quantity of cooling water used was 60.5 lbs. per H.P. of blowing cylinder, while the heat carried off by the cooling water amounted to 16% of the heat in the gas. These too are very excellent results. The quantity of cylinder oil used in the power cylinder was 1.19 lbs. (about .16 gallon) per hour. The other parts to be lubricated received per hour 2.72 lbs. (about .37 gallon) of fresh oil besides a certain quantity of recovered oil.

5. Louis Soest & Co. in Reisholz-Düsseldorf. (Plate XXIII.)

As a result of its excellent constructive features (compare the main parts shown in Figs. 87–89, 144–147, and 276 in Part II) this new engine has made a name for itself in very short time. One of the two 600-H.P. twin engines shown on Plate XXIII was exhibited in Düsseldorf in 1903. The two-throw crank-shaft, forged from one solid block, weighed 32100 lbs., of which 5700 lbs. belonged to the balance weights. The diameter of the wheel, for a coefficient of regulation of $\frac{1}{150}$th was 19.7 ft. ($V \sim 125$ ft. per sec.). The weight of the wheel was 66000 lbs., corresponding to 110 lbs. per rated horse-power. The indicator card, Fig. 519, was obtained when operating on a blast-furnace gas carrying about .0006 gram of dust per cubic foot and having a heating value of 95.5 B.T.U. per cu.ft. The mean effective pressure of this diagram was about 71 lbs. per sq.in., corresponding to a horse-power developed of 660 B.H.P.

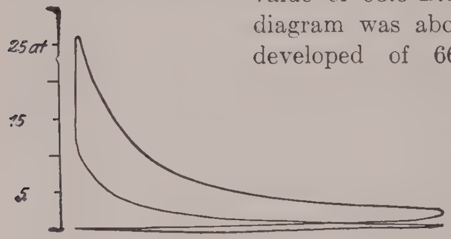


FIG. 519.

Late in 1903 this firm took up the manufacture of double-acting 4-cycle engines for the largest sizes and is at present engaged in developing a double-acting 2-cycle machine (similar to the Körting, but having the firm's own method of forming the mixture and of governing). A 400-H.P. machine of this type is already being used for experimental work.

6. Güldner-Motoren-Gesellschaft in Aschaffenburg. (See Plates XXIV and XXV).

The degree to which the efficiency and specific capacity of a given type of internal-combustion engine may be raised by intelligent treatment of the constructive features is well shown in the new Güldner engine. Fundamentally nothing but a simple 4-cycle engine, this machine owes the unusual efficiency and capacity shown in the tables below entirely to careful attention to all its details, and especially to the design of the compression and combustion chamber, and the mixing, ignition, and governing arrangements. Fig. 520, together with Plate XXIV, show the main constructive features of one of the 100-H.P. engines. Probably the most striking feature is the double wall A-frame, which is a development of the author's own older designs for the first Diesel engine, operating without a cross-head and which of late has become well known in the newer Augsburg design of this type of machine.

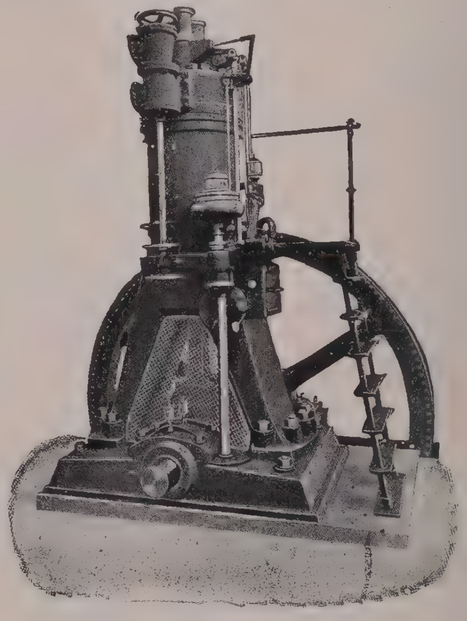


FIG. 520.—100 H.P. Güldner Engine.
($D=18.7''$, $S=27.5''$, $n=160$.)

For reasons strongly emphasized in all parts of this book, the Güldner engine at all times operates with a mixture as pure and as uniform as possible and in every respect brought to the best condition attainable for most efficient combustion. One means to this end is the temporary combined action of the exhaust and the inlet valve such that the combustion chamber is filled with air even before the suction stroke proper is commenced. It follows that the cylinder will get a greater quantity of cool and pure mixture, which, at all loads on the engine, will ignite with certainty and burn efficiently. (See the developed regulation diagram, Fig. 521.) The speed of the engine is regulated by means of the governor and gear shown in Figs. 327 and 328, p. 245. This arrangement operates to proportion the quantity of charge to the load;

but to obviate as far as possible the difficulty encountered in igniting the mixtures at the lower loads, brought about by lower compression pressures, the governor also controls the proportion of gas in the charge. The success of this system of governing explains why the heat consumption per horse-power-hour of these engines shows such a comparatively slow increase with decrease of load.

The erecting plan, Figs. 522 and 523, shows the practical disposition of the apparatus making up a suction-gas plant, arranged with a view to accessibility in all its parts. Attention is also called to the short lengths and accessibility throughout of the gas main, important points in themselves. In the larger installations the exhaust pipe is jacketed and the jacket is used directly to conduct away the cylinder cooling water.

Whenever possible the exhaust muffler is placed near the roof either in the engine or in the producer room, the idea being to promote ventilation by thus heating the upper layers of the air. For the same purpose, the inlet of the suction pipe to the engine is placed in the producer room. (For details of Güldner producers, see p. 280.)

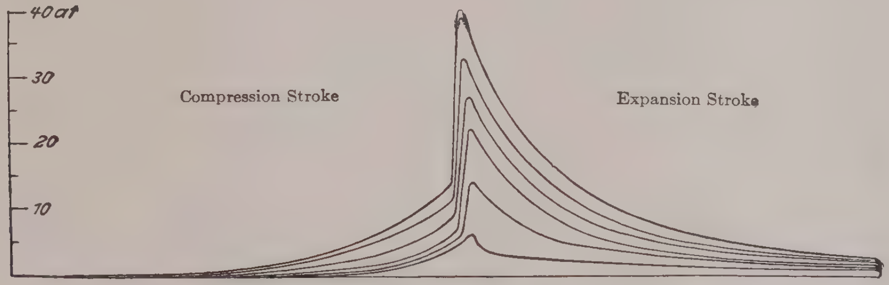


Fig. 521.—Regulation Diagram, Max. Load to No Load.

The gas and air controlling valves of the large Güldner machines may be operated from the floor as well as from the platform. In these sizes the exhaust valve is also water cooled. All of the sizes made have ignition gear that may be set by hand, pressure oil pumps for the pistons, and a central oiling system for the bearings. Up to 20 H.P. the starting is accomplished by means of the starting gear shown in Figs. 443 and 444, p. 291; the larger machines are started by compressed air obtained with the small compressors illustrated in Figs. 451–453, p. 294. The details of some of the other machine parts are shown in some of the type constructions illustrated in Part II.

At present Güldner engines are built as single cylinder engines with capacities ranging from 10 to 125 H.P., and two cylinder units are made in sizes from 100 to 250 H.P. The main dimensions of the engines as well as the room required under ordinary conditions, are shown in Table 56 following:

TABLE 56

PRINCIPAL DIMENSIONS OF SINGLE-CYLINDER GÜLDNER SUCTION GAS PLANTS

(See Plans, Figs. 522, 523)

N_n in B.H.P.		10	12	15	20	25	30	35	40	50	60	70	80	100	125
R.p.m.		240	240	225	225	210	210	200	200	190	180	170	160	155	150
Main dimensions of engines, producers, and room required, in ft.	*A*	7.87	8.20	9.20	9.85	9.85	10.5	10.5	11.80	12.75	13.75	14.75	15.70	18.0	21.30
	B	3.28	3.40	4.28	4.60	4.93	5.25	5.91	6.33	7.07	7.87	8.55	9.18	9.84	12.10
	C	4.60	4.93	5.58	5.91	5.91	6.23	6.57	6.90	7.22	7.57	8.20	8.53	9.52	10.50
	D *	10.50	10.50	11.80	13.10	13.75	15.10	16.40	16.40	16.40	17.40	19.00	20.35	21.30	22.30
	E	6.92	7.20	7.70	8.05	8.53	8.85	9.85	9.85	9.85	10.50	11.50	12.50	13.10	14.40
	F	1.13	1.65	2.62	2.78	2.78	3.45	3.12	4.43	5.08	6.08	5.25	6.23	6.90	8.85
	G	6.55	6.55	6.55	7.05	7.05	7.05	7.35	7.35	7.70	7.70	9.50	9.50	11.15	11.50
	H	5.29	5.29	5.29	6.27	6.27	6.27	7.06	7.06	8.05	8.05	9.85	9.85	11.50	13.75
	I	1.97	1.97	1.97	1.97	1.97	1.97	1.97	1.97	1.97	1.97	1.97	1.97	2.62	2.62
	K	2.62	2.62	2.62	2.96	2.96	2.96	3.28	3.28	3.61	3.61	3.94	3.94	4.27	4.58
	L	1.64	1.64	1.64	1.64	1.64	1.64	1.64	1.64	1.64	1.64	1.64	1.64	1.97	1.97
	M	5.08	5.24	5.58	5.58	5.90	5.90	6.25	6.25	6.55	6.88	7.55	8.20	9.50	11.15
	N	3.22	3.28	3.44	3.72	3.77	3.94	4.03	4.27	4.59	5.08	5.37	5.63	6.07	6.55
	O	5.57	5.90	6.56	7.38	7.87	8.53	9.50	9.50	9.85	10.80	11.15	11.80	11.80	12.15
	P	.36	.38	.39	.39	.41	.43	.44	.46	.73	.79	.86	.92	.98	1.05
	Q	1.97	1.97	2.21	2.46	2.95	3.28	3.62	3.77	3.94	4.27	4.58	5.24	5.73	6.23
	R	.92	.92	1.07	1.15	1.31	1.37	1.44	1.51	1.80	2.10	2.39	2.52	2.95	3.28
Width of belt		.43	.43	.46	.53	.59	.62	.66	.69	.82	.95	1.08	1.15	1.34	1.51

* This amount of head-room is ample to allow of the taking out of the piston together with the connecting-rod by means of a crane. Where this room is not available the piston may be taken out by itself, in which case the head-room may be made equal to about $\frac{3}{4}D$.

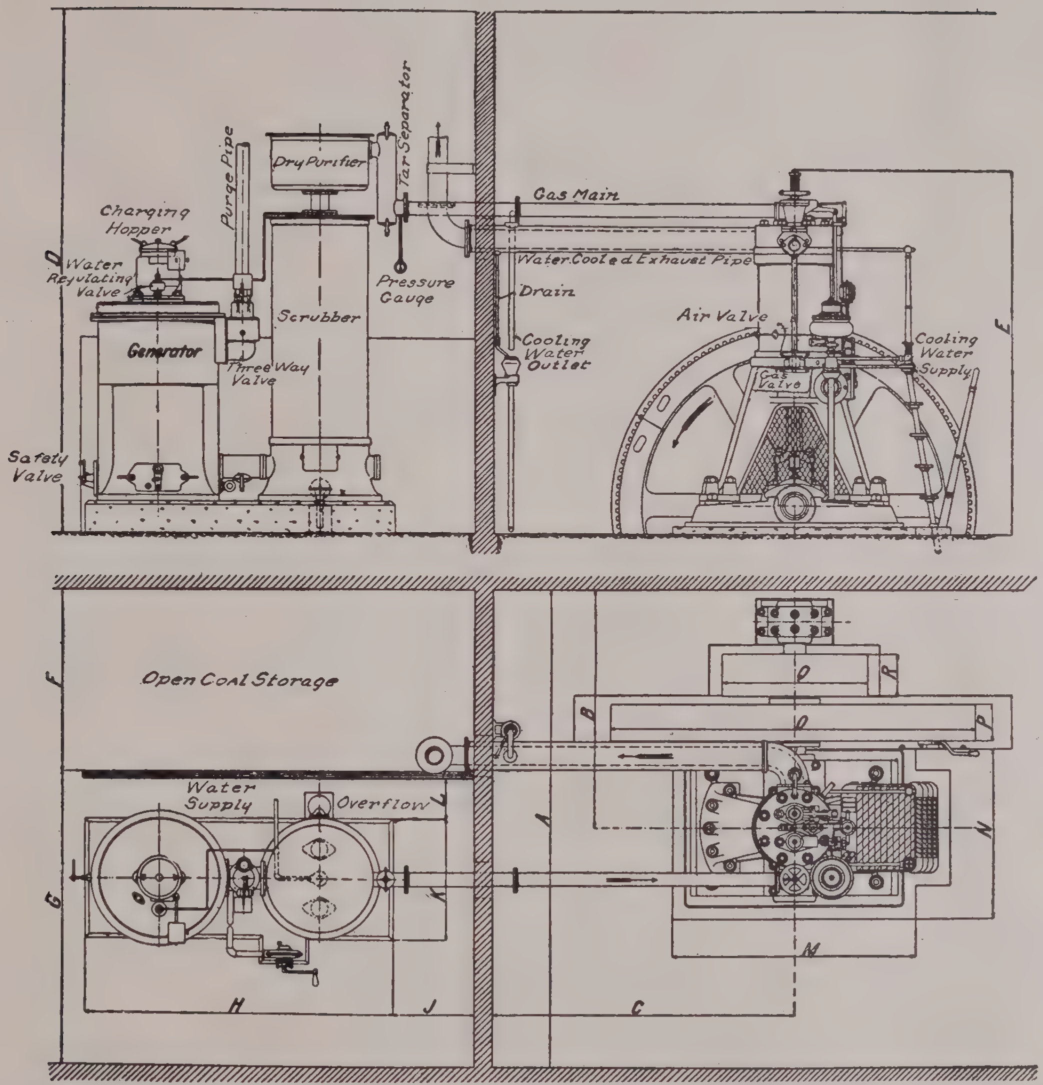
Operating Results. (*a*) The first vertical Güldner engine in actual operation was tested in 1903 by Prof. M. Schröter with the assistance of Dr. Koob. The tests were conducted with both illuminating and anthracite suction gas. The engine was rated at 20 H.P., cylinder diameter = 9.87″, stroke = 15.75″. The maximum compression was 113.5 lbs. with illuminating gas and 157 lbs. with suction gas. In both cases the indicator diagrams showed a mean effective pressure in excess of 113 lbs. per square inch at maximum load. The exhaust gases at all loads were invisible and odorless.

TABLE 57

RESULTS OF TESTS WITH ILLUMINATING GAS

Approximate Load.	Mean R.p.m.	Mean Effective Pressure, lbs. per sq.in.	Average I.H.P.	Lower Heating Value of Gas, B.T.U. per cu.ft.	Consumption of Gas Referred to 32° F. and 735.5 mm. Hg.		Consumption of Gas Reduced to a Heating Value of 560 B.T.U. per cu.ft.		Heat used per I.H.P. Hour, B.T.U.	Indicated Thermal Efficiency, η_i, %
					Per Hour, cu.ft.	Per I.H.P. Hour, cu.ft.	Per Hour.	Per I.H.P. Hour.		
Half maximum load	211.8	59.5	19.1	488	299	15.60	275	14.40	8640	31.6
Same	213.9	63.7	21.7	496	295	14.25	279	13.45	7550	33.9
¾ maximum load	212.8	95.3	30.9	495	384	12.43	362	11.72	6580	38.8
Same	213.7	94.0	30.6	501	383	12.38	360	11.80	6620	38.6
Nearly maximum load	214.5	114.5	37.1	497	457	12.28	436	11.73	6590	39.0
Same	210.7	110.0	35.3	498	399	11.25	377	10.65	5970	42.7

The heat balance for the last line of the above table reads as follows: Of the heat supplied in the illuminating gas, 42.7% are transformed into indicated work, 33.2% are



FIGS. 522 and 523.—Erecting- and Floor-plan for Single-cylinder Güldner Suction-gas Plants. (See Table 56 for Dimensions.)

lost in the cooling water, while 24.1% (remainder) are lost in exhaust and by radiation.

The same high thermal efficiency is also shown when operating with suction gas. As Table 58 shows, the first machine to be operated on this gas showed an indicated thermal efficiency of nearly 29%, which includes all losses in producer and scrubber. This corresponds to a heat consumption of from 6750 to 7150 B.T.U. per I.H.P. per hour for the engine alone, which would, in round numbers, be equal to .55 lb. of anthracite per I.H.P., a figure which was afterwards found in actual service (see p. 359).

In Table 58, "Total" anthracite consumption includes coal used for starting, etc. "Net" consumption indicates the coal actually used per horse-power. The coal used (brand Kohlscheid III) contained 85.18% C, 3.49% H, 5.36% ash, 1.97% H_2O, and had a mean heating value of 14 000 B.T.U. per lb.

TABLE 58

RESULTS OF TESTS WITH SUCTION GAS

Approximate Load.	R.p.m.	Mean Effective Pressure, lbs. per sq.in.	Mean I.H.P.	Heating Value of Anthracite, B.T.U. per lb.	Anthracite used per I.H P. Hour, $H_u = 14\,000$ B.T.U per lb		$H_u = 14\,400$ B.T.U. per lb.		Heat Consumed in Producer per I H P. Hour.	
					Total, lbs.	Net lbs.	Total, lbs.	Net, lbs.	Total, B.T U.	Net, B T.U.
Nearly maximum	210	108	34.4	14000	.747	.637	.727	.617	10450	8830

(*b*) While Tables 57 and 58 show the results referred to the indicated horse-power, and therefore give some idea of the progress made by Güldner engines with regard to thermal efficiency, the tables following show the results of acceptance tests made by independent observers and consequently the information refers more particularly to the economic side.

TABLE 59

ANTHRACITE CONSUMPTION OF TWO 30-H.P. GÜLDNER ENGINES UNDER PARTIAL LOADS

(Determined in the Municipal Lighting Plants of Mosbach and Niederbronn)

Test No.	Duration of Test (No Interruption). Hours.	Min.	Approximate Load.	Mean R.p.m.	Efficiency of Dynamo and Belt, %	Average Load on Engine. B.H.P.	I.H.P.	Mechanical Efficiency η_m, %	Fuel used per Hour, including Coal for Starting, lbs. per K.W.	B.H.P.	I.H.P.	Fuel used per Hour, excluding Coal for Starting, lbs. per K.W.	B.H.P.	I.H.P.
I	9	20	normal	220.8	85.3	35.1	44.2	.80	1.425	.885	.703	1.22	.746	.602
II	8	25	$\frac{3}{4}$	220.6	85.0	25.9	37.4	.70	1.74	1.0	.754	1.43	.897	.618
III	8	20	$\frac{2}{3}$	221.4	82.0	21.5	31.1	.69	1.77	1.10	.842	1.56	.945	.740
IV	8	30	$\frac{1}{2}$	221.9	78.9	17.65	26.4	.67	2.93	1.42	.945	1.97	1.13	.753
V	6	50	$\frac{1}{3}$	223.0	72.0	13.30	22.2	.60	3.51	1.86	1.12	2.83	1.48	.894
VI	7	35	$\frac{1}{4}$	222.6	70.2	11.15	18.85	.59	4.34	2.23	1.33	3.36	1.74	1.03

Maximum horse-power developed continuously by each engine, 45 B.H.P., for short periods 50–51 B.H.P. Kind of anthracite used in Tests I, II, IV, and VI (Niederbronn): No. III Nut, Förtingsiepen, heating value average 13 500 B.T.U. per lb.; kind used in Tests III and V (Mosbach), No. III Nut, Herstal (Belgium), average heating value 13 860 B.T.U. per lb. Efficiency of belt transmission assumed =97% in all cases, efficiency of dynamo determined from calibration curves of manufacturer. The fuel consumption is *not* recomputed to the basis of 14 400 B.T.U. coal.

TABLE 60

VARIATION IN SPEED IN A 30-H.P. GÜLDNER ENGINE IN THE MUNICIPAL LIGHTING PLANT AT NIEDERBRONN UNDER SUDDEN CHANGES OF LOAD

Test No.	Sudden Change of Load. from K.W.	to K.W.	% of Rated Load.	Speed Varied. from R.p.m.	to R.p.m	% of Normal Speed.	Remarks.
I	22.5	15.6	−32.1	220	223	+1.50	The speed variations were observed by means of tachometer.
II	13.4	22.5	+42.4	225	222	−1.44	
III	17.8	8.8	−42	221	225	+1.68	Weight of fly-wheel appr. 5500 lbs., diameter 9.25 ft. Coefficient of regulation at normal load $\delta = \frac{1}{80}$.
IV	18.0	0	−84	221	229	+3.50	
V	0	20.4	+95	226	221	−2.48	

Table 59 shows a comparatively slow increase of the fuel consumption per H.P. with decreasing load and also proves that uninterrupted continuous operation is possible even for the smallest loads. The latter fact, as well known, depends mainly upon the effectiveness of the producer end of the installation.

The rapid accomodation of the gas producers to any calls made upon them in combination with the excellent speed regulation of the engine are also well brought out in Table 60, the figures for which are also taken from the above mentioned acceptance test.

(*c*) The above mentioned Niederbronn plant tested by two Dutch engineers, Van Burkom and Huibers, after it had been in operation one year, at the direction of the management of the electric station at Scheveningen, gave the following results in everyday operation:

Producer lighted at	7:15 A.M.
Engine started (first ignitions)	7:25
Time required to start installation	10 min.
Full load on engine at	7:29
End of test, no interruptions	5: P.M.
Time under full load	9 hrs, 31 min.
Total coal consumed, including that used for starting up, lost in ash, etc.	244 lbs.
Power developed, from wattmeter	177.2 K.W. hours.
Anthracite consumed, including coal for starting, etc., per effective K.W. hour	1.375 lbs.

In order to be able to determine the net coal consumption, the fuel used between 10:30 A.M. and 5 P.M., together with the electric output, were separately recorded. These figures gave the following results:

Net coal used 10:30 to 5:00	138.5 lbs.
Electrical output for same period	111.3 K.W. hours.
Hence net coal consumed per effective K.W. hour	1.255 lbs.
net coal consumed per I.H.P. hour, approximate	.539 lbs.

These results include all losses due to belt transmission and dynamo up to the switchboard. Coal was anthracite, Langenbrahm mine, Size II, which was used as received.

The load on the engine during the test was kept as constant as variations in the power demand in the outside lines would allow. Toward the end the load was gradually reduced in order to let the producer burn down as far as possible. At times the mean effective pressure as per indicator card exceeded 100 lbs. per square inch.

During the entire 9½ hours, the producer was cleaned but once (at 2:30 in the afternoon), but the load was not changed, so that no interruption in the regular operation occurred.

To complete the capacity and economy tests, a few measurements were also taken on the speed variations under sudden changes of load. This showed, for instance, that with a sudden decrease from about 35 B.H.P. to no load, the speed rose from 212 to 220 r.p.m, which is less than 4%.

After the completion of the tests the mixing and inlet valve was taken out for examination. The surfaces showed but very light deposits, not enough to interfere in any way with operation, although at that time the valve had been in use over fourteen days, from 12–14 hours each day. An examination of the cylinder walls and piston face also disclosed but slight deposits. According to the records of the station, the piston had at no time been taken out of the cylinder for purposes of cleaning, etc., since the engine was erected.

(*d*) The electric station in Bergzabern, Pfalz, until 1904 operated by steam, in that year installed a 30-H.P. Güldner suction gas plant. The comparative statement of the fuel cost for the two systems is of interest.

TABLE 61

FUEL COST OF STEAM PLANT AND SUCTION GAS PLANT IN MUNICIPAL STATION AT BERGZABERN

No.	Fuel Consumption and Fuel Costs.	Suction Gas Plant.		Steam Plant.	
		Total.	Net.	Total.	Net.
1	Coal consumption per B.H.P. hour, lbs	.848	.720	6.92	6.47
2	Coal consumption per Kilowatt hour, lbs.	1.475	1.260	12.60	11.70
3	Cost of coal per B.H.P. hour, cents	.315	.266	1.32	1.22
4	Cost of coal per Kilowatt hour, cents.	.512	.467	2.39	2.20

The figures for the gas plant were determined by direct measurement, the figures for the steam plant were computed from the average as shown by older operating records of the plant. The difference between total and net consumption, that is, between the consumption with and without starting losses, amounted to approximately 15% in the gas plant and 9.3% in the steam engine. At the place of consumption the anthracite used cost $7.35 per ton, and steam coal $3.72 per ton.

The ratio between the fuel costs in this case is approximately 4.5 to 1 in favor of the gas plant. In the electric stations at Mosbach and Niederbronn, which formerly also operated with steam engines, the ratio is about the same (3 or 4 to 1). It must of course be admitted that the suction gas plant cannot show any such saving when it comes to compete with up-to-date and fully loaded steam plants, but even in such cases the economic superiority of gas over steam is certain. As far as the older steam plants are concerned, there is little doubt that the substitution of suction gas installations for steam is about the only successful means of making the operation pay. The above examples from practice certainly show that.

The economy figures just quoted are all the more remarkable owing to the fact that they were obtained with comparatively low compression pressures. While most suction-gas engines operate with compression from 180 to 225 lbs.,[1] some going even higher than that, the Güldner engine at normal load compresses to 135 lbs., and only compresses to from 165 to 180 lbs. when working at maximum load with a charge larger than normal. The indicator diagrams, Figs. 524 and 525, bring out this point, and also show the unusually large overload capacity of the engine. The mean effective pressure for maximum load exceeds 130 lbs. per square inch. In order to get a basis of comparison, it may be stated that in the ordinary suction-gas engine, the mean effective pressure is usually from 75 to 90 lbs. per square inch (see Fig. 485), and that only the Diesel oil engine shows anywhere near the maximum mean effective pressure indicated in Fig. 524. (In modern steam engines, at normal load, the mean effective

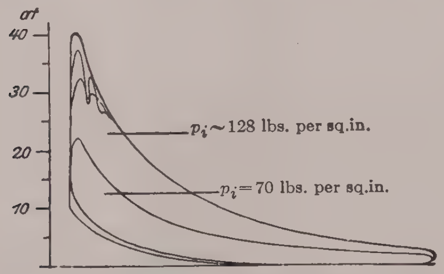


FIG. 524.—Diagrams obtained at Max. Load and near Normal Load.

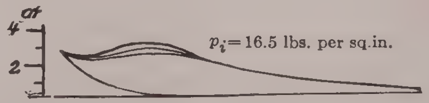


FIG. 525.—No Load Diagram, smallest Degree of Compression (driving only belt and counter-shaft.)

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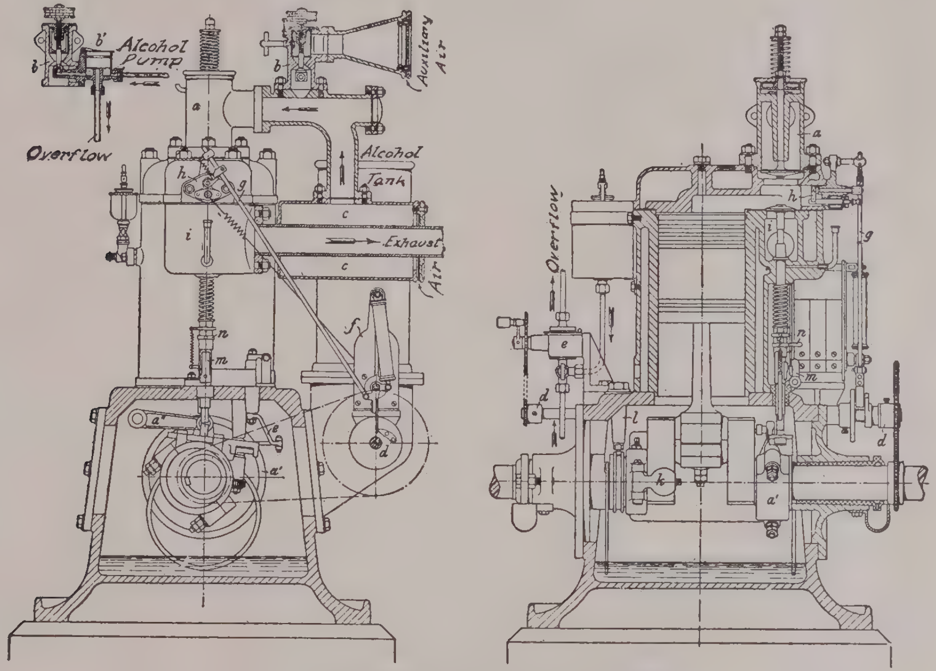
[1] TRANSLATOR'S NOTE. These figures seem high for American practice.

pressure varies from 30 to 40 lbs., which is approximately $\frac{1}{4}$ of that shown by internal-combustion engines. On this basis the Güldner engine, in spite of the 4-cycle operation, is about on a par with the ordinary double-acting steam engine as far as specific capacity is concerned.)

7. Motorenfabrik Oberursel A. G., in Oberursel.



Figs. 526 and 527.—"Gnom" Alcohol Engine.

Inlet valve *a* is automatic, ahead of this valve is placed atomizer *b* with overflow *b'*. At the same time that the main quantity of air is drawn through the pre-heater *c*, an auxiliary quantity is furnished through the atomizing valve. Secondary shaft *d* is used to operate alcohol pump *e*, magneto *f*, and make-and-break mechanism *g* for the igniter *h*. Exhaust valve *i* is controlled by the combined gear *a'* and *a''*. Shaft governor *k*, by means of the rod *l* and the latch *m*, holds open the exhaust valve when the speed of the engine exceeds certain limits. The shift link *n* is used to reduce compression at starting.

Table 62

DIMENSIONS AND WEIGHTS OF "GNOM" ENGINES

N_n in B.H.P.	1	2	3	4	5	6	8	10	12	15	20	25
R.p.m.	400	360	350	300	300	290	280	270	260	250	250	180
Length ....ft.	2.30	2.62	2.79	3.11	3.45	3.45	3.77	3.94	4.27	4.27	4.92	5.58
Width* ....ft.	3.28	3.77	3.94	4.43	4.60	5.43	5.91	6.57	6.38	7.38	7.23	6.57
Height ....ft.	3.12	3.45	3.62	3.94	4.27	4.43	5.09	5.58	6.08	6.08	7.23	6.72
Net weight ...lbs.	925	1515	1700	2240	2310	3380	4400	4980	6150	7250	8340	10550
Total weight..lbs.	1320	1925	2200	2840	2990	4180	5280	5820	7250	8460	9450	11450

* Parallel to shaft.

The engine is sold under the trade name of "Gnom." It may be arranged to operate also on kerosene or on illuminating gas by substituting proper mixing arrangements for the atomizer. The original designer of this machine is Hugo Seck, Sr., who apparently has borrowed the idea of the enclosed crank case and splash lubrication from American steam-engine practice.

8. Vereit. Maschinenfabrik Augsburg und Maschinenbau-Ges. Nürnberg A.-G., Werk Augsburg. (See Plates XXVI to XXVIII.)

This firm is the leading and by far the most important manufacturer of Diesel engines. In place of the first model, which had a cross-head, the type at present built operates without a cross-head (Fig. 528 and Plate XXVI) and being cheaper, finds a readier sale. Other licensed firms, however, notably the Maschinenfabrik L. A. Riedinger in Augsburg, retain the cross-head construction, which undoubtedly gives a more satisfactory machine.

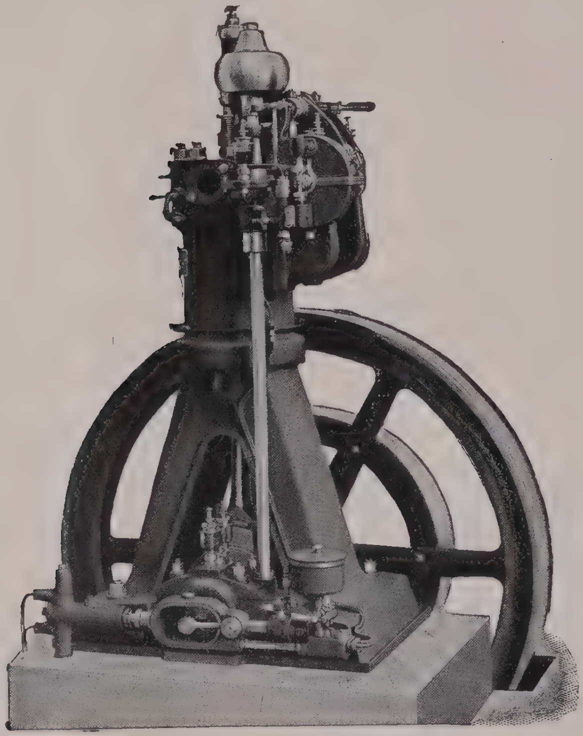


Fig. 528.—Type of small Diesel Engine. (Rated B.H.P. = 8 at 270 r.p.m.)

In Plate XXVI, *a* is the inlet, *b* the exhaust, *c* the fuel injection, and *d* the starting valve, all operated by cams fixed on the lay shaft. The levers of the last two valves are fastened to an eccentric sleeve. By turning the latter through 90°, the two levers mentioned are shifted from the starting position (in which the fuel injection valve is out of commission) to the normal operating position (see left figure in Plate XXVI). The starting of the engine is now done by operating on the 4- instead of the 2-cycle principle. The high pressure air is obtained in two stages, pump *e* through valve *d* taking a small quantity of air, preliminarily compressed, from the main cylinder near the end of the compression stroke, compressing it from about 150 to 650 or 700 lbs. per square inch, and storing it in the starting or injection tanks. Oil pump *f*, whose suction valve is controlled by the governor, forces a quantity of oil in direct proportion to the load into the nozzle of the injection valve *c*. Piston and wrist-pin lubrication is taken care of by the small pump *g* with its oil reservoir *g'*.

The operation of the engine is readily understood. During the first down stroke the cylinder is filled with air through the inlet valve *a*. This air is next compressed to about 480 lbs., during which it is heated to from 1000 to 1100° F. Shortly before the piston reaches the upper dead center, the fuel-injection valve *c* opens, and the oil which has been furnished by the oil pump *f*, is injected in a fine spray into the highly heated charge of air in the cylinder by means of high pressure air obtained as above described. The oil burns as it enters the cylinder, being ignited by the highly heated air which it meets. The expansion caused by the heat thus developed forces the power piston downward on the third stroke of the cycle, the pressure remaining approximately constant until after about $\frac{1}{8}$ stroke at normal load the fuel valve *c* closes. The gases then expand behind the piston to the end of the stroke. During the next and last stroke of the cycle the burned gases are expelled through valve *b* and the engine is then ready for the next charge. It therefore operates on the 4-cycle principle.

In the smaller commercial machines, Fig. 528, the small air pump is attached to the side of the base plate at the height of the crank-shaft, and is directly driven from the end of this shaft. The small oil pump and the governor belonging to these machines are shown in Fig. 405, p. 267.

TABLE 63

DIMENSIONS AND WEIGHTS OF DIESEL ENGINES MADE BY THE MASCHINENFABRIK AUGSBURG

	SINGLE-CYLINDER ENGINES.													
N_n B.H.P.	4	6	8	10	12	15	20	25	30	35	40	50	60	70
R.p.m.	290	280	270	255	250	235	215	205	195	190	180	170	165	160
Dimensions over all:														
Across shaft . . . . ft.	4.27	4.93	5.25	5.90	6.56	7.22	7.55	8.20	8.53	8.85	9.50	9.85	10.15	10.50
Parallel to shaft . . . . ft.	3.28	3.94	3.94	4.59	4.59	6.56	7.05	7.37	7.87	8.53	9.18	9.85	10.50	11.15
Height above floor . . . . . . ft.	4.58	5.57	5.57	6.56	6.56	7.22	7.87	8.53	8.85	9.50	10.15	10.80	11.50	12.45
Head room required, with reference to dismounting . ft.	8.53	9.17	9.17	9.50	9.50	12.15	13.10	13.80	14.10	14.45	15.10	16.10	17.05	18.05
Depth of foundation . . . . . . ft.	3.28	3.28	3.94	4.58	5.23	5.90	6.23	6.56	6.88	7.21	7.53	7.86	8.20	8.53
Diameter of fly-wheel . . . . . ft.	4.27	4.92	5.23	5.90	6.56	7.21	7.53	8.20	8.52	8.85	9.50	9.85	10.15	10.50
Net weight . . . . lbs.	1980	2640	3190	3850	4520	7140	9680	11900	14300	16900	19600	24200	29700	35200
Total weight . . . . lbs.	2420	3190	3740	4620	5280	8580	11450	14300	17200	20200	23300	28600	35200	41800

	SINGLE-CYLINDER ENGINES.		TWIN-CYLINDER ENGINES.										
N_n B.H.P.	80	100	30	40	50	60	70	80	100	120	140	160	200
R.p.m.	160	160	235	215	205	195	190	180	170	165	160	160	160
Dimensions over all:													
Across shaft . . . . ft.	11.15	11.50	7.21	7.54	8.21	8.53	8.85	9.50	9.85	10.15	10.50	11.15	11.50
Parallel to shaft . . . . ft.	11.80	12.45	9.20	9.85	10.80	11.15	11.80	12.45	13.10	14.10	15.10	15.70	16.40
Height above floor . . . . . . ft.	13.45	14.45	7.20	7.86	8.53	8.85	9.50	10.15	10.80	11.50	12.45	13.45	14.45
Head room required, with reference to dismounting . ft.	19.70	21.35	12.45	13.10	13.80	14.10	14.45	15.10	16.05	17.05	18.05	19.70	21.35
Depth of foundation . . . . . . ft.	8.85	9.20	5.91	6.23	6.56	6.87	7.21	7.53	7.86	8.20	8.52	8.85	9.17
Diameter of fly-wheel . . . . . ft.	11.15	11.50	7.21	7.53	8.20	8.53	8.85	9.50	9.85	10.15	10.80	11.15	11.50
Net weight . . . . lbs.	40700	50600	12800	16900	21100	25500	30100	35200	44000	54800	63800	72600	91300
Total weight . . . . lbs.	47200	59400	15400	20200	25300	30800	35800	41800	51600	63800	74800	84600	105500

Operating Results with the Diesel engines made by the Maschinenfabrik Augsburg. (*a*) Tests made by Prof. E. Meyer, in September, 1900, on a 30-H.P. motor of the older type (with cross-head) but compressing the injection air in two stages, as in later types: Principal dimensions of the working cylinder, $D = 11.8''$, $S = 18.25''$; of the air pump, $D = 1.97''$, $S = 3.15''$; ratio of compression, $\varepsilon = 16.3$. The first six tests recorded in Table 64 were made with American kerosene, $\gamma = .796$ at 64.4° F., the last four with Tegernseer crude oil, $\gamma = .789$ at 63.4° F.

TABLE 64

R.p.m.	177.4	181.1	182.6	184.0	183.3	185.8	181.6	181.2	181.8	185.0
Brake horse-power	38.90	29.8	30.0	23.45	15.05	8.41	29.8	29.75	23.18	15.18
Mean effective pressure in power cylinder, lbs. per sq.in.	106.0	85.4	85.4	70.3	53.2	37.9	85.7	88.2	71.0	55.9
Mean indicated friction resistance, p_r, lbs. per sq.in.	19.25	20.0	20.0	19.85	20.7	20.0	—	—	—	—
Indicated horse-power *	47.50	38.90	39.20	32.60	24.70	17.75	39.20	40.30	32.55	26.00
Mechanical efficiency, %	81.9	76.3	76.3	71.9	61.0	47.3	76.0	73.7	71.2	58.3
Oil used per I.H.P. hour . . . lbs.	.402	.348	.346	.348	.350	.366	.357	.350	.352	.335
Oil used per B.H.P. hour . . . lbs.	.482	.457	.454	.483	.576	.764	.469	.475	.496	.575

* Without deduction for air pump horse-power, but less an allowance equivalent to 1.4 lbs. per sq.in. for suction and exhaust resistances.

Assuming that the heating value of the oil is $H_u = 18\,550$ B.T.U. per lb., the above figures show the indicated thermal efficiency at full load to be $\eta_i =$ approximately 39% and the economic efficiency $\eta_w =$ approximately 30%.

(*b*) Following tests (*a*), Prof. Meyer also determined the fuel consumption of a Diesel engine of the older type, which had been in operation in a textile mill since the beginning of 1899. The engine was of the two-cylinder type, 60 B.H.P., with cross-head and single-stage air pump. It received no special preparation for the test. The oil consumption amounted to .407 lb. per I.H.P. hour at 68.3 I.H.P. (after subtracting pump work). The oil used was Tegernseer crude ($\gamma = .8125$ at 62.6° F.). Estimating the mechanical efficiency at $\eta_m = 69.5\%$, this means a consumption of .513 lb. of oil per B.H.P. hour and an economic efficiency of approximately 26%.

(*c*) The thermal efficiency of the newer design of Diesel engine, without cross-head, is shown by a series of tests made by Prof. E. Meyer in June, 1902, on two engines of 70 and 8 B.H.P. respectively. The details of the 70 B.H.P. engine are shown in Plate XXVI, those of the 8 B.H.P. machine in Fig. 528. Table 65 contains the important results of the tests. The engine dimensions were:

$$N_n = 70 \text{ B.H.P.} \begin{cases} \text{Power cyl. } D = 15.75'', & S = 23.60'', \quad \varepsilon = 15.4 \\ \text{Pump cyl. } D = \ \ 2.21'', & S = \ \ 5.47'', \quad \text{Air pressure at beg.} = 120\text{–}150 \text{ lbs.} \end{cases}$$

$$N_n = \ \ 8 \text{ B.H.P.} \begin{cases} \text{Power cyl. } D = \ \ 6.48'', & S = 10.70'', \quad \varepsilon = ? \\ \text{Pump cyl. } D = \ \ \ .59'', & S = \ \ 1.97'', \quad \text{Air pressure at beg.} = 120\text{–}150 \text{ lbs.} \end{cases}$$

The fuel used for the first five tests on the large engine was Russian kerosene ($\gamma = .806$ at 65.5°), for the rest of the trials the fuel was a paraffin oil of dirty-brown color ($\gamma = .893$ at 59°). The small engine was tested only with the Russian kerosene.

The heating value of the kerosene was afterward determined with great accuracy at 18 550 B.T.U. per pound, consequently for the tests showing the best oil consumption in the table the thermal efficiencies figure out as follows:

	70 B.H.P. Engine.	8 B.H.P. Engine.
Indicated thermal efficiency, η_i	.402	.353
Economic efficiency, η_w	.326	.280

Both of these engines obtained their air for injection purposes by preliminary compression in the main cylinder, as above explained. The scheme has certain disadvantages, for which reason several of the licensed firms have already abandoned its use.

Table 65

	70 B.H.P. Engine.								
Revolutions per minute	157.9	158.8	158.7	159.8	160.5	157.8	159.0	159.3	159.9
B.H.P.	85.3	68.6	68.55	52.2	34.40	85.25	68.70	68.85	52.25
Mean effective pressure power cylinder, lbs. per sq.in.	116.50	96.20	96.50	79.00	57.60	122.00	101.00	101.80	82.80
Total I.H.P.	107.5	89.2	89.3	73.5	54.0	112.3	94.1	95.0	77.5
Mean effective pressure, air pump, lbs. per sq.in.	304.00	277.00	274.00	295.50	226.75	288.00	309.50	315.00	316.50
Indicated pump H.P. (estimated for the small engine)	2.53	2.32	2.29	2.48	1.93	2.40	2.60	2.67	2.68
Net I.H.P.	105.2	86.9	87.1	71.0	52.0	109.9	91.5	92.3	74.9
Mechanical efficiency %	81.4	79.1	78.9	73.6	66.2	77.7	75.2	74.8	70.0
Oil used per I.H.P. hour lbs.	.342	.339	.339	.328	.331	.361	.339	.344	.335
Oil used per B.H.P. hour lbs.	.420	.430	.432	.450	.500	.467	.455	.460	.479
Cooling water per hour (inlet temp. = 51.8° F.) lbs.	—	1275	1945	925	308	2240	1540	1495	1715
Temperature of cooling water, outlet ° F.	152	162–178	155–158	158–170	160–173	171–158	165–168	155–175	171–175
Exhaust temp. (measured in pipe just beyond valve) ° F.	874	637–660	644–660	635–637	462–457	787–792	652–682	627–719	610–627

	8 B.H.P. Engine.					
Revolutions per minute	267.1	278.4	270.3	274.7	276.3	284
B.H.P.	9.91	8.75	8.49	6.13	4.61	0
Mean effective pressure power cylinder, lbs. per sq.in.	107.50	92.80	94.30	75.15	61.60	26.95
Total I.H.P.	12.76	11.50	11.32	9.20	7.55	3.40
Mean effective pressure, air pump, lbs. per sq.in.	—	—	—	—	—	—
Indicated pump H.P. (estimated)	.30	.30	.30	.30	.25	.20
Net I.H.P.	12.46	11.20	11.02	8.90	7.30	3.20
Mechanical efficiency %	79.4	78.1	77.0	69.0	63.2	—
Oil used per I.H.P. hour lbs.	.389	.397	.384	.344	.366	—
Oil used per B.H.P. hour lbs.	.489	.507	.495	.522	.581	—
Cooling water per hour (inlet temp. = 51.8° F.) lbs.	216	297	—	—	—	—
Temperature of cooling water, outlet ° F.	167	145	—	—	—	—
Exhaust temp. (measured in pipe just beyond valve) ° F.	—	—	—	—	—	—

(*d*) Worthy of attention, also, are a series of tachometer measurements which the Maschinenfabrik Augsburg itself has made on an 80-H.P. two-cylinder engine, having cross-head and single-stage air pump. The dimensions of the engine were: $D = 11.80''$, stroke $= 18.10''$, $n = 190$ r.p.m. It was direct connected to a dynamo and had two fly-wheels, the weight of each of which was 5720 lbs., with an effective radius of 3.68 ft. The computed coefficient of regulation at full load was 1% ($\delta = .01$). The governor was adjusted to regulate to within 4%. The measurements were taken with a Horn tachograph, the diagram showing the following results:

At the normal load the tachograph indicated a value of $\delta = \frac{1}{45}$ when one wheel was in use, and a value of $\delta = \frac{1}{80}$ when both wheels were in place. Both values, however, are larger than the value of δ computed on the basis of fly-wheel weight. This is probably due to inaccuracies in the tachograph diagrams, owing to inertia effects in the moving parts of the instrument itself. The voltmeter for all changes of load, whether one or both wheels were used, did not show a variation of more than $\frac{1}{2}$ volt. A

25 candle-power incandescent lamp showed a slightly unsteady light when the engine was running at about normal load with one wheel; the light, however, was absolutely steady when two wheels were employed.

TABLE 66

SPEED VARIATION IN AN 80-H.P. DIESEL ENGINE

ONLY ONE WHEEL IN USE.				BOTH WHEELS IN USE.			
Sudden Change of Load		Per Cent of Normal Load.	Speed Variation, Per Cent.	Sudden Change of Load		Per Cent of Normal Load.	Speed Variation, Per Cent.
from K.W.	to K.W.			from K.W.	to K.W.		
24.2	42.9	+48.0	−2.2	41.7	23.8	−46.0	+1.73
43.2	24.2	−48.8	+2.25	24.0	16.75	−18.6	+ .51
24.2	16.9	−18.7	+1.0	16.75	8.35	−21.5	+ .60
16.9	8.14	−22.5	+1.15	16.5	23.7	+18.5	− .75
8.14	24.2	+41.2	−2.0	23.7	42.2	+47.5	−2.0
24.2	43.2	+48.8	−2.4	42.2	16.5	−66.1	+2.4
43.2	16.9	−67.5	+2.95	16.5	41.8	+64.9	−2.1
				41.8	7.9	−87.0	+3.75

TABLE 67

FUEL CONSUMPTION OF DIESEL ENGINES PER B.H.P. HOUR, HEATING VALUE OF THE OIL ASSUMED 18 000 B.T.U. PER POUND

(Guarantee figures furnished by the Maschinenfabrik Augsburg, reserving a permissible fluctuation of 10%.)

Single-cylinder engine	H.P.	8	10	12	15	20	25	30	35
Two-cylinder engine	H.P.	. .	. .	. .	30	40	50	60	70
From normal to full load	lbs.	.517	.507	.485	.473	.462	.452	.440	.430
	B.T.U.	9300	9120	8730	8520	8300	8120	7920	7750
Three-quarter load	lbs.	.538	.528	.507	.495	.485	.473	.462	.452
	B.T.U.	9700	9500	9120	8920	8730	8520	8300	8120
One-half load	lbs.	.627	.616	.594	.573	.560	.550	.538	.528
	B.T.U.	11250	11080	10690	10300	10080	9900	9700	9500
One-quarter load	lbs.	.858	.825	.792	.770	.737	.727	.715	.705
	B.T.U.	15400	14830	14250	13850	13250	13050	12880	12690

Single-cylinder engine	H.P.	40	50	60	70	80	100	125
Two-cylinder engine	H.P.	80	100	120	140	160	200	250
From normal to full load	lbs.	.430	.430	.418	.407	.407	.407	.407
	B.T.U.	7750	7750	7530	7320	7320	7320	7320
Three-quarter load	lbs.	.452	.452	.440	.430	.430	.430	.430
	B.T.U.	8120	8120	7920	7750	7750	7750	7750
One-half load	lbs.	.517	.517	.507	.495	.495	.495	.495
	B.T.U.	9300	9300	9120	8920	8920	8920	8920
One-quarter load	lbs.	.694	.683	.672	.660	.660	.660	.660
	B.T.U.	12500	12300	12080	11880	11880	11880	11880

III. Other Types of Foreign Gas Engines

1. Langen & Wolf in Wien (Vienna). This firm is the Austrian licensee of the Gasmotoren-Fabrik Deutz, but has in the past few years brought out a construction of its own, the design of H. Ebbs, one of the partners of the firm. An external view of this machine is shown in Fig. 529.

The engine has a pleasing appearance. The construction is especially distinguished by the two distance pieces, before the first cylinder and between the cylinders, and the method of tying together frame, cylinders, and distance pieces by four continuous tie-rods or bolts. The latter are intended to insure central alignment of the various parts and to relieve the cast-iron distance pieces of tensile stresses as far as possible.

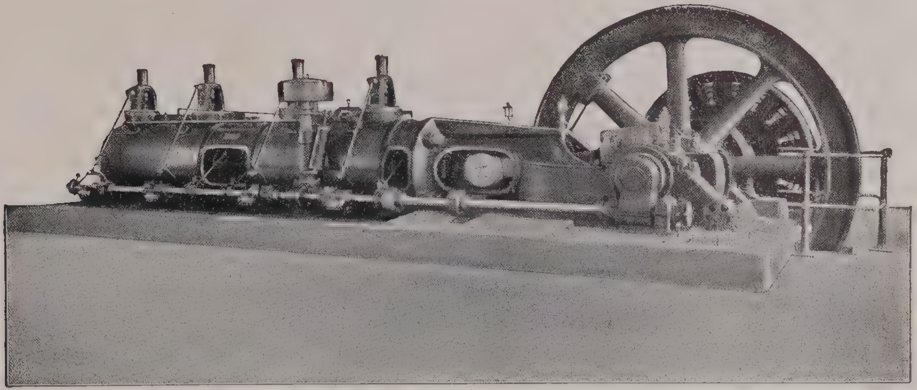


Fig. 529.—General View of a 300 H.P Double-acting Tandem Engine, Langen & Wolf, Vienna.

Further details of construction are given by the cross-section, Fig. 530, p. 368. The back cylinder cover is easily dismounted and is made of such diameter that the piston may be taken out at that end without trouble. The jacket wall is cast in one piece with the cylinder, and in this connection the division of the wall at the middle plane *a* for the purpose of relieving temperature stresses is worthy of attention. The engine is intended to operate on producer gas, but to make possible the use of illuminating gas, where the latter may be used as a reserve or as the starting medium, the inlet valve housings are supplied with suitable connecting flanges for this purpose. For information regarding valve gear and regulations, see Fig. 277, p. 209.

2. Société anon. John Cockerill in Seraing. (See also Plates XXIX and XXX.) The engines built by this company all operate on the ordinary 4-cycle principle, and therefore need no very extended description. The various types at present brought out are well illustrated in the following figures and plates. (Of special interest is the new valve gear employing eccentrics, for details of which see Figs. 278–282.)

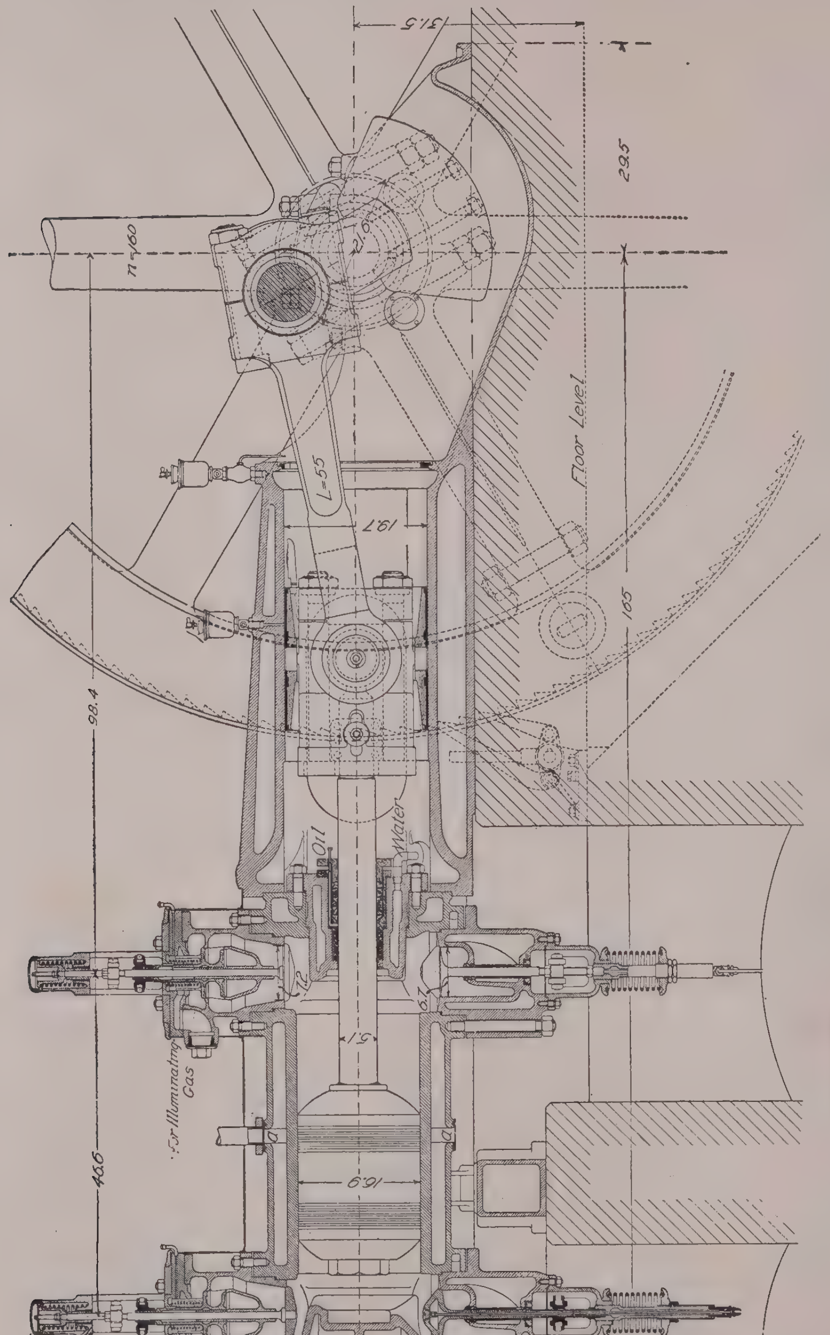


FIG. 530.—Cross-section of a 100–120 H.P. Double-acting 4-cycle Engine. Langen & Wolf, Vienna. (For Valve Gear, see Fig. 277.)

Operating Results.[1] (*a*) Tests made Witz and Hubert, 1898, on one of the older 200-H.P. simplex engines built by the Cockerill Co. for blast-furnace gas. The cylinder diameter was 31.5″, the stroke 39.3″. The test lasted twenty-four hours:

R.p.m.	105.2
Explosions per minute	47.6
B.H.P.	179.00
I.H.P., approximate	212.00
Mechanical efficiency	85%
Gas used per B.H.P. hour	119.1 cu.ft.
Higher heating value of gas	110 B.T.U.
Water used in the washers, per cu.ft of gas	19.8 lbs.
per B.H.P. hour	67.0 lbs.
Cooling water per B.H.P. hour	93.3 lbs.
Temperature cooling water, inlet	62.8° F.
outlet	93.6° F.
Total water consumption per B.H.P. hour	160.3 lbs.
Cylinder oil per B.H.P. hour	.033 lb.
Grease for bearings per B.H.P. hour	.0042 lb.

The compression was 107 lbs. per sq.in., the mean effective pressure 52.5 lbs. per sq.in. Gas consumption and heating value are referred to 46.4° F. and 760 mm. Hg.

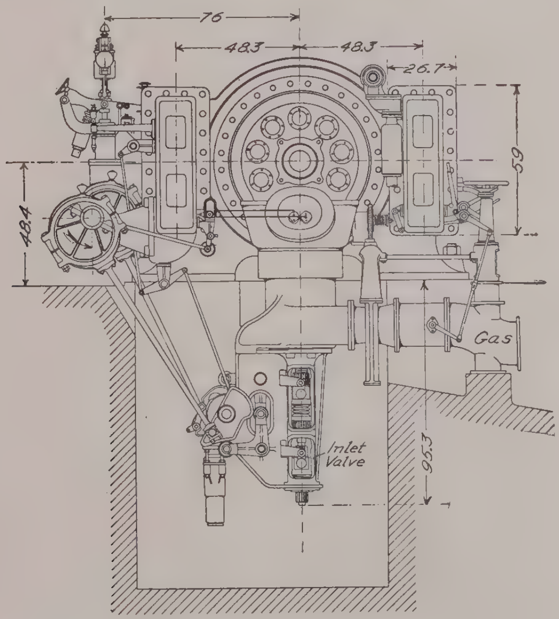


Fig. 531.—End View, Cockerill Engine. (Belongs to Plate XXIX.)

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[1] Translator's Note. The tests quoted under (*a*) and (*b*) are given in the historical part of Güldner's Handbook, not translated in this volume. They are quoted here to show the results attained with the early Simplex engines, designed by Delamare-Deboutteville, and built by the Cockerill Co.

(*b*) Tests made by Hubert early in 1900, on a single-acting 600-H.P. simplex blast-furnace gas engine built by the Cockerill Co. Cylinder diameter 51.2″, stroke 55″:

TEST I. ENGINE WITHOUT BLOWING CYLINDER (6 hours)

R.p.m.	94.57
Explosions per minute	41.90
I.H.P.	775.0
B.H.P.	567.0
Mechanical efficiency, %	73.14
Gas used per B.H.P. hour, cu.ft.	124.8
Heating value of gas, B.T.U. per cu.ft.	110.5
Temperature of gas at cylinder, °F.	48.2
Temperature of exhaust gas, °F.	948.0
Cooling water used per B.H.P. hour: In cylinder jacket, lbs.	126.50
In piston, lbs.	28.30
Total cooling water, per B.H.P. hour	154.80
Temperature cooling water, inlet, °F.	46.15
outlet, °F.	91.70

TESTS II AND III. BLOWING CYLINDER IN OPERATION. (Each test 2 hours)

	II.	III.
R.p.m.	83.92	93.02
I.H.P., engine	736.00	873.00
I.H.P. in blowing cylinder	555.0	715.00
Total mechanical efficiency	75.33	81.81
Gas consumption per B.H.P. hour, cu.ft.	112.0	101.8
Heating value of gas, av. B.T.U. per cu.ft.	111.2	112.7
Gas temperature at cylinder, °F.	46.75	49.80
Temperature of exhaust gas, °F.	912	1017
Cooling water per B.H.P. hour: Cyl. jacket, lbs.	115.5	89.3
Piston, lbs.	29.9	22.7
Total, lbs	145.4	112.0
Temperature increase of cooling water, °F.	82.10	87.60

The average heat balance was as follows: 28% transformed into work, 52% lost in cooling water, and 20% lost in exhaust and by radiation. The compression pressure was 135 lbs. per square inch, the mean effective pressure 67.4 lbs. per square inch, the explosion pressure from 248 to 284 lbs. The heating values of the gas above given is referred to 46.4° F. and 760 mm. Hg., and was determined by Witz in a bomb calorimeter. Tests with Junker's calorimeter gave only 88, 102.5, and 100 B.T.U. per cubic foot, that is, approximately 10% less.

(*c*) In addition to the above the following table shows a series of results obtained by Prof. Hubert in 1901 on tests of a 200-H.P. Cockerill engine:[1]

PRINCIPAL DIMENSIONS

Cylinder diameter . . . . . . 33.45″, Stroke	39.40″
Normal r.p.m.	100
Diameter of fly-wheel, feet	13.10
Peripheral velocity of wheel ft. per sec.	69.0
Weight of wheel, lbs.	23000
Fly-wheel weight per H.P. approximate, lbs.	165.0

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[1] The original report published in the Revue univ. des mines, etc., 1902, p. 273, contains numerous diagrams and tachographs.

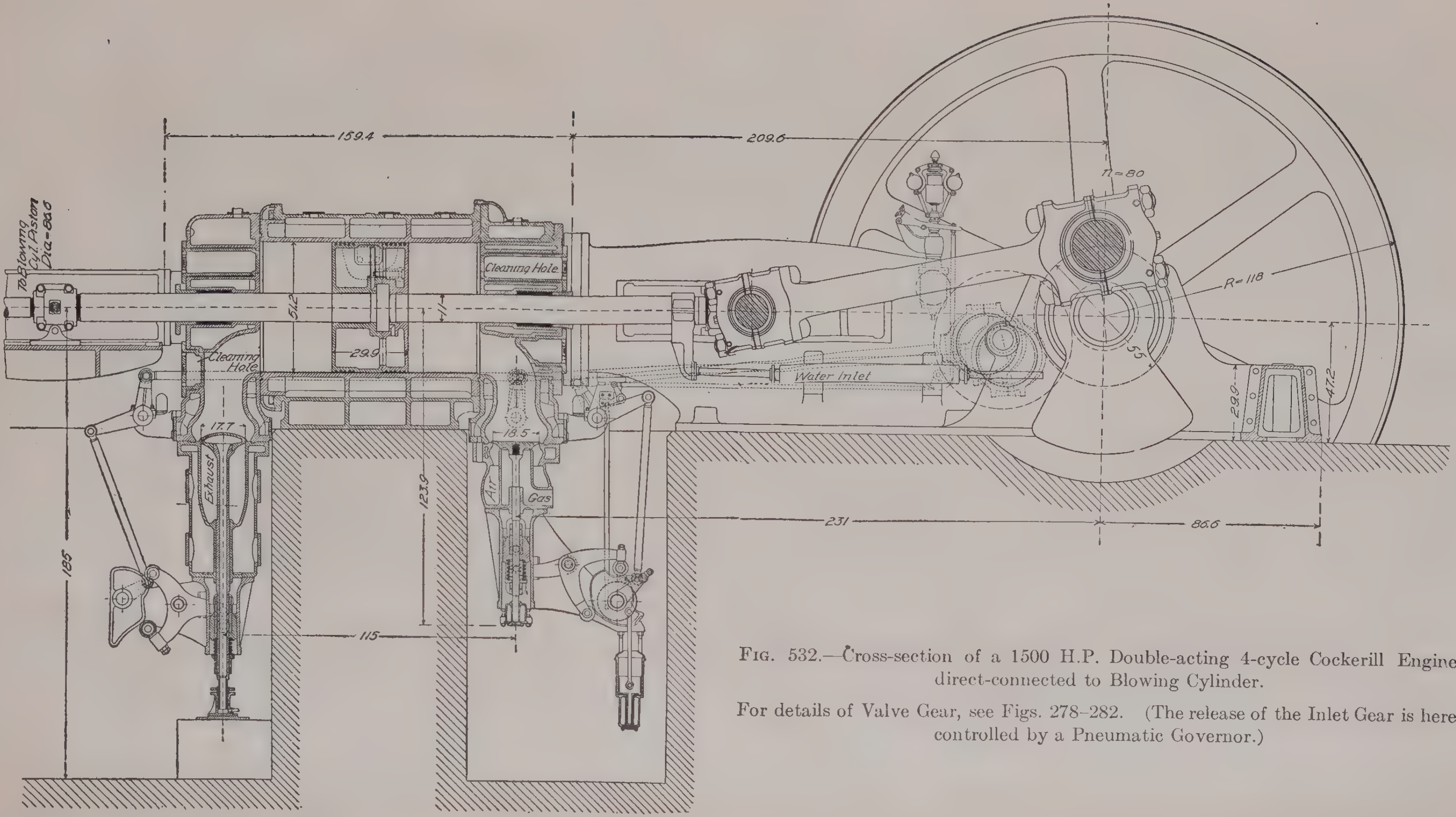

FIG. 532.—Cross-section of a 1500 H.P. Double-acting 4-cycle Cockerill Engine direct-connected to Blowing Cylinder.

For details of Valve Gear, see Figs. 278–282. (The release of the Inlet Gear is here controlled by a Pneumatic Governor.)

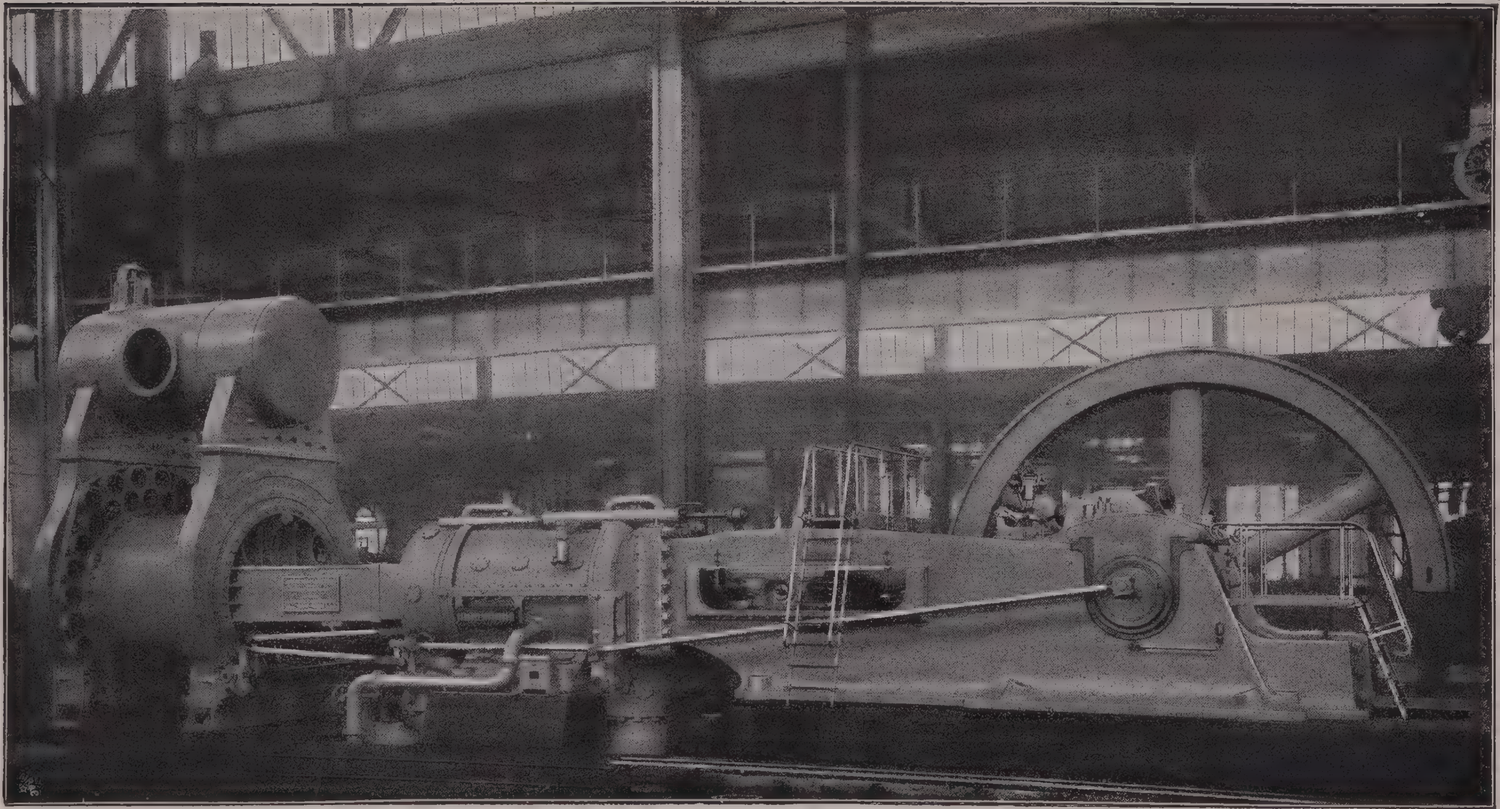

FIG. 533.—General View of a Double-acting 4-Cycle Cockerill Engine direct-connected to Blowing Cylinder. Rated H.P. = 1500 at 80 r.p.m. ($D = 51.2''$, $S = 54.2''$.)

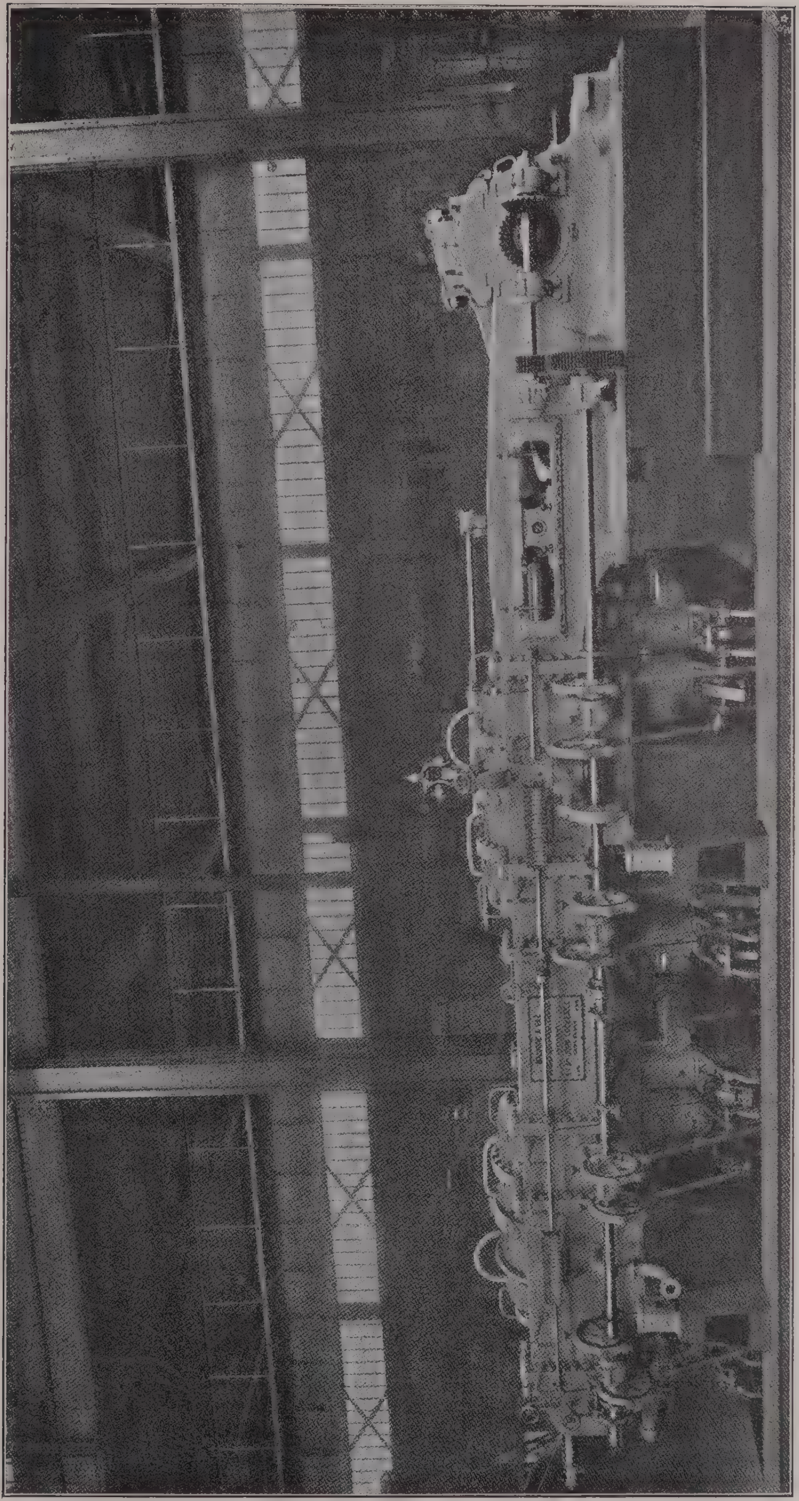

FIG. 534.—General View of a Double-acting Tandem Cockerill Engine. Rated H.P. = 300 at 80 r.p.m. ($D = 51.2''$, $S = 54.2''$.)

Table 68

Test No.	Load.	R.p.m.	I.H.P.	B.H.P.	Mechanical Efficiency, %.	Heating Value of Gas, B.T.U. per cu.ft.	Gas Used per		Heat Supplied per		Indicated Thermal Efficiency, %.	Economic Efficiency, %.	Heat in Cooling Water, %.	Temperature of Exhaust Gas, Deg. F.
							I.H.P. Hour, cu.ft.	B.H.P. Hour, cu.ft.	I.H.P. Hour, B.T.U.	B.H.P. Hour, B.T.U.				
1	no load	101.35	45.5	0	....	108.9	247.5	....	27000	....	9.4	....	30.0	915
2	¼	103.20	90.0	63.0	70.2	116.0	174.5	247.0	20900	28620	12.6	8.9	22.1	955
3	½	98.56	152.5	121.9	80.0	118.2	115.5	144.5	13640	17050	18.6	14.9	29.4	985
4	¾	98.90	208.5	173.0	82.9	113.0	97.3	117.2	11000	13250	23.2	19.1	26.7	1042
5	full	98.60	224.0	196.9	87.6	113.8	115.0	131.3	13100	14950	19.5	17.0	30.2	1192
6	full	98.60	242.5	218.0	90.0	137.2	108.7	120.8	12440	13750	20.5	18.5	30.0	1164
7	no load	105.90	46.8	0	—	110.2	244.5	—	27000	—	9.4	—	47.4	916
8	¼	106.35	106.2	56.5	53.3	101.7	141.2	265.0	14320	26900	17.8	9.5	36.1	962
9	½	100.62	150.5	110.8	73.8	96.2	107.5	146.0	10230	13950	24.8	18.2	44.0	995
10	¾	100.37	206.0	167.8	81.5	98.9	107.5	131.9	10980	13080	23.2	19.3	31.3	1037
11	full	98.18	233.0	209.0	90.0	109.5	106.5	119.0	11670	12980	21.9	19.6	30.9	1092
12	no load	101.37	57.0	0	—	103.8	270.0	—	28000	—	9.1	—	26.3	785
13	¼	101.33	100.3	53.7	53.6	100.8	179.5	334.0	18050	33700	14.1	7.6	28.3	924
14	½	100.90	149.5	111.0	74.5	101.1	136.0	182.0	13900	17630	18.4	14.5	26.0	970
15	¾	100.00	202.3	167.0	82.8	96.4	114.0	137.1	11200	13300	23.1	19.1	27.7	1029
16	full	99.64	238.0	212.0	89.2	108.9	110.2	124.0	12100	13440	21.3	18.9	23.3	1092
17	full	99.74	244.0	212.3	87.2	104.1	106.5	122.2	11100	12720	23.0	20.0	24.2	1107

For tests Nos. 1 to 11 the engine was regulated by means of a pneumatic governor; for the last six tests an ordinary centrifugal governor was used to control the speed. With either governor the speed variation under sudden increase or decrease of load varying between normal load and no load was from 4 to 7%, the total speed variation between these two limits of load being on the average 10%. Within one 4-stroke cycle, the angular velocity at normal loads varied from 3 to 4%, at the maximum load from 5 to 6%. The average compression and explosion pressures were respectively: At ¼ load, 55.5 and 105 lbs. per square inch; at ½ load, 61.2 and 156 lbs. per square inch; at ¾ load, 119 and 191 lbs. per square inch; and at maximum load, 153.5 and 239 lbs. per square inch.

3. Compagnie "Duplex" pour la fabrication des moteurs à gas in Paris. One of the first French engine builders to use the double-acting 4-cycle engine for large capacities was Niel, the designer of the "Duplex" engine. The machine does not offer much of anything new regarding construction, but employs a method of charging which is quite different from that used in other engines. The charge is drawn in on one side of the piston and is distributed between the two cylinder ends, so that each end works with only about one-half full charge for the purpose of obtaining complete expansion.

As shown in Figs. 535–536,[1] each cylinder end has an inlet valve *b* and an exhaust valve *a*. The two inlet valves are connected by a passage or port *c*, which at the middle carries the mixing valve *d*, which in turn obtains air through the valve *e* and gas through valve *f*. All the valves are operated by cams from a lay shaft. The operation is as follows: In Fig. 536, as the piston starts to the right it draws in a full charge behind it, at the same time forcing the exhaust gas out of the right-hand end of the cylinder. On the return stroke one-half of the charge in the left-hand end is forced through the port *c* and the inlet valves into the right-hand end of the cylinder, the valves *d*, *e*, and *f* having closed. At the middle of the stroke, both valves *b* also close. Then follows compression in the left end and expansion of the charge in the right end of the cylinder. The charge in the left end is ignited by hot tube at the proper time. The piston now moving to the right compresses the mixture in that end, this is in turn ignited when the piston reaches the crank end. The two ignitions are therefore 180° apart, while exhaust and charging actions occupy the other 540° of the cycle. When the speed exceeds the normal, the governor temporarily prevents the opening of the gas valve *f*.

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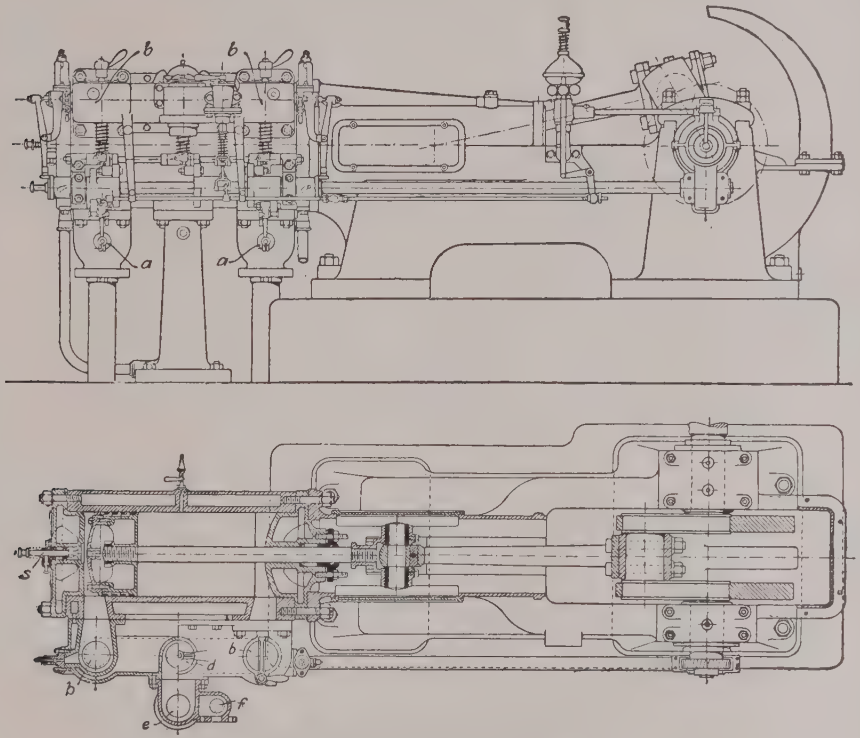
[1] Z. d. V. D. I., 1901, p. 325.



FIGS. 535 and 536.—Duplex Engine.

The indicator cards taken when operating with illuminating gas show that the compression is about 90 lbs. and the explosion pressure varies from 300 to 450 lbs. The mean effective pressure is only about 43 to 50 lbs. per square inch. The terminal pressure is close to atmosphere, so that the exhaust is practically noiseless. Definite test figures for this engine are apparently not available, which does not speak particularly well for its economy.

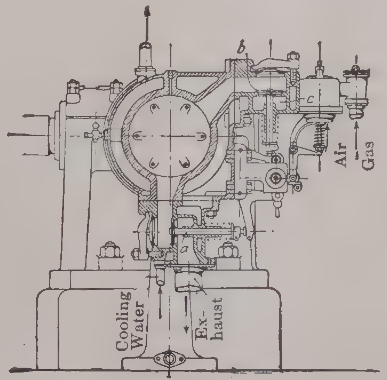
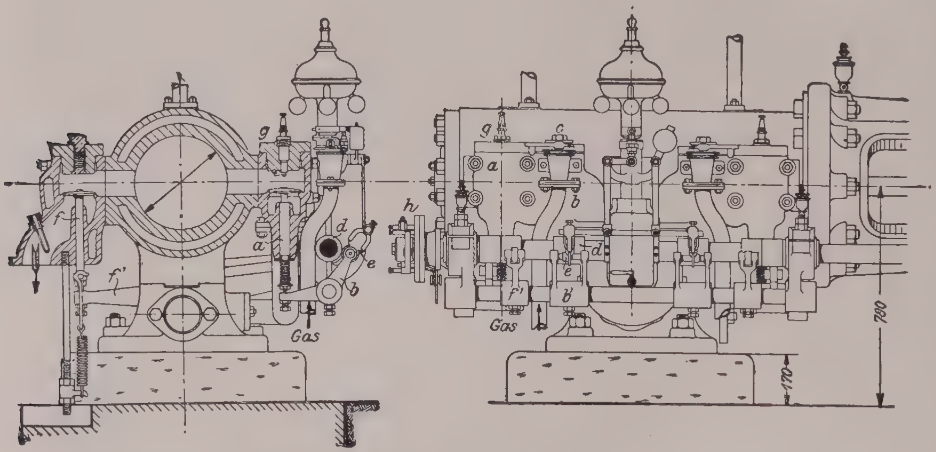


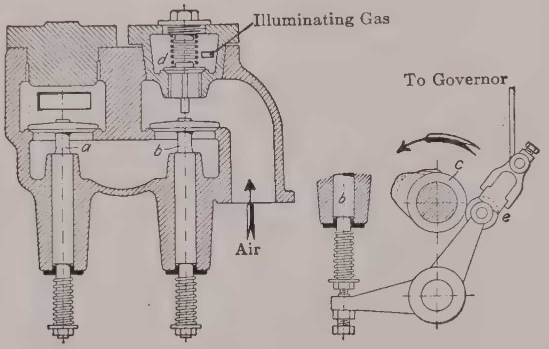
FIG. 537.

4. Société anon. d'exploitation des brevets Letombe in Lille. This engine, which received a great deal of attention at the last world's fair at Paris, is distinguished by a very promising system of governing, which regulates both quantity and quality of charge. In contradistinction to other methods of governing, the quantity of the charge decreases and the gas content increases as the load increases, that is, the higher the load, the smaller the total volume of charge, but the richer the mixture. On the other hand, in accord with reasons based on thermal considerations, the leaner mixtures at the lower loads receive the higher compression.



Figs. 538 and 539.—Letombe Engine.

For larger capacities, this engine is built as a double-acting 4-cycle, Figs. 538 and 539. The most important details of its design are found in the arrangement and operation of the air and gas valves. The main inlet valve *a*, Figs. 540 and 541, is automatic, that is, it acts as a simple check or suction valve. The governing valve *b*, placed just ahead of *a*, is actuated by the step cam *c*, the valve stem of *b* also lifting the gas valve *d*. The governor controls the position of the roller *e*, so that it runs on one of the four steps of the cam, depending upon the load. Valve *b* is opened at the same time for all loads, but the moment of closing and the effective lift of this valve are so regulated that, if the load is decreasing, the valve closes later and the lift is



Figs. 540 and 541.

smaller than it was before. Since gas valve d is directly operated by b, this valve is affected in the same way. The last step of the cone is so low that the gas valve d is not lifted at all, in which case the engine will miss the next cycle.

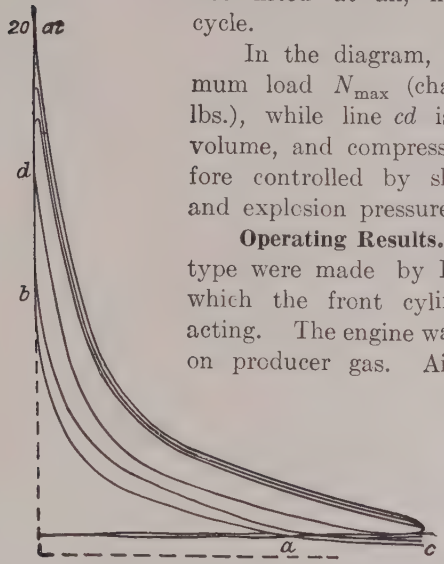


Fig. 542.

In the diagram, Fig. 542, line ab is the compression line for maximum load N_{max} (charge = 50–55% cylinder volume, compression = 120 lbs.), while line cd is the compression line for N_{min} (charge = cylinder volume, and compression = 195 lbs.). The work area developed is therefore controlled by shifting the compression line, while expansion line and explosion pressure remain approximately the same.

Operating Results. The first accurate tests of an engine of this type were made by Prof. Witz in July, 1902, on a tandem engine, in which the front cylinder was single-acting, the back cylinder double-acting. The engine was rated at 300 B.H.P. and was intended to operate on producer gas. Air and steam were forced through producer and the gas made through scrubber to a gas holder by means of a Root blower, which was operated by a 3-H.P. auxiliary engine. The cylinder diameter of the main engine was 23.65″, the stroke was 31.5″. The piston rod between the two cylinders had a diameter of 7.08″. The coal used was anthracite containing 14.8% of volatile matter and about 3% of ash. The test lasted eleven hours, with an interruption of one hour, and gave the following average results:

Table 69

R.p.m.	135.56
Total B.H.P.	289.00
H.P. in blower engine	3.00
Net B.H.P.	286.00
Anthracite per B.H.P. hour, lbs.	.830
Heating value of gas per bomb calorimeter, B.T.U. per cu.ft.	148.00
Temperature of cooling water, °F.	131–149
Temperature of exhaust gas, °F.	658
Average compression pressure, lbs. per sq.in.	118

The economy is apparently very good, but no better than is attained by other first class machines regulating by means of one or the other of the common systems of governing. Compare, for instance, pp. 328 and 358.

5. Premier Gas Engine Co., Nottingham. The Premier gas engine, introduced during the last few years for large capacities, apparently with great success, is of the *4-cycle positive scavenging type,* that is, near the end of exhaust stroke the combustion space is thoroughly flushed out with fresh air in order to receive a fresh cool charge and to cool down the interior cylinder and piston walls.[1] Accurate drawings of the constructive details of the engine were not obtainable, but the method of operation and the general features of the design may be made clear by Fig. 543, which represents a 600-H.P. tandem engine.[1] All Premier engines less than 300 B.H.P. have only one cylinder.

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[1] This method of scavenging a 4-cycle cylinder before the new charge comes in is not a new idea, and was already in use by Daimler in 1885 in some of his small engines, but the Premier Co. first applied it to large engines.

Air pump *a*, beginning with the crank position shown in the figure (about .8 of the exhaust stroke), scavenges alternately the combustion spaces *b* and *c*, until the end of the stroke. The air supply in each case amounts to about 3½ times the volume of the space to be cleared out, and passes through ports *d* and *d′*, and inlet valves *e* and *e′* respectively. At the end of the scavenging period in the cylinder concerned, the suction stroke next follows as in ordinary operation. The power pistons *b* and *c* are connected by means of side rods *f*, which on one end are held by the yoke *g*, while on the other they are connected to the pump piston *a*. The rods thus pass through the annular space of the pump *a*, but not through the combustion space of the cylinder *b*. Grid valve *h* is operated by rod and eccentric from the main shaft, and not only serves to control the suction and discharge passages of the pump *a*, but also at the proper time establishes communication between the outer air and the passage *d* through which one or the other of the power cylinders may then draw its charge of air. During the first third of the instroke, pump *a* forces a part of its charge of air back through the suction ports, for the rest of the stroke up to the beginning of scavenging the air is compressed into the ports *d* and *d′*. Each inlet valve housing *e* is furnished with a special piston valve which keeps the gas ports closed during scavenging, but opens the air ports wide. During the suction stroke this valve then opens the gas ports and closes the air ports partly until the right ratio of air to gas is obtained. When the speed becomes too high the governor throws the operating rods of the gas valve temporarily out of action. The gas ports then remain closed and the cylinder draws nothing but air. The heads of the power piston as well as the exhaust valves are water-cooled. Ignition is effected by two open hot tubes in each cylinder.

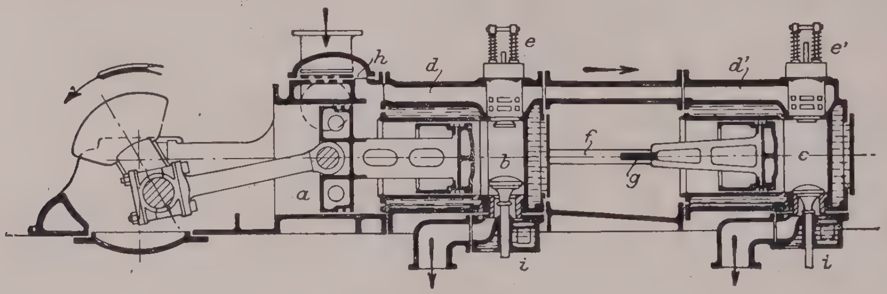

FIG. 543.—Premier Engine.

Operating Results. (*a*) Professor Humphrey obtained the following results on a 600-H.P. tandem engine operating on Mond gas. The machine had a cylinder diameter of 28″, a stroke of 30″, while the speed was 128 r.p.m.

$$\left.\begin{array}{lr}\text{Total indicated horse-power,} & 489\\ \text{Indicated pump horse-power,} & 34\end{array}\right\}\text{Net I.H.P., 455}$$

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[1] Engineering, 1901, p. 195.

Brake horse-power, 386.0
Total mechanical efficiency, $\eta_m = 75.7\%$, net $\eta_m = 81.0\%$.
Heating value of Mond gas, approximate, 144 B.T.U. per cu.ft. $\Big\}\ \eta_i = 31.6\%$
Gas used per B.H.P. hour, 69.2 cu.ft. $\quad \eta_w = 25.6\%$

In this case both cylinders worked at about $\frac{2}{3}$ load each.

(*b*) A second test on the same engine, made by loading up the back cylinder and cutting out the front cylinder, gave results as follows:

$n = 127.4$ r.p.m., $N_i = 320.92$ I.H.P., $N_e = 216$ B.H.P., $\eta_m = 67.1\%$.
Heating value of gas, 139.7 B.T.U. per cu.ft., gas consumption per B.H.P. hour, 71.8 cu.ft., $\eta_i = 37.76\%$, $\eta_w = 25.3\%$.

In this test the heat carried off by the cooling water per hour was distributed as follows:

Total heat in cooling water		667 000 B.T.U. = 100%
Carried off in	Cylinder jacket	467 000 B.T.U. = 70%
	Piston	119 000 B.T.U. = 18%
	Exhaust valve	81 000 B.T.U. = 12%

(*c*) Finally, the same test showed that at a maximum load equal to 650 I.H.P., the pump work was 36 I.H.P. = 5.5% of the I.H.P. of the power cylinder, the mechanical efficiency was $\eta_m = 83.8\%$ (including pump work), or $\eta_m = 88.8\%$ (excluding pump work).

During the first tests the indicator diagrams, with an explosion pressure of barely 340 lbs. per square inch, showed a mean effective pressure of approximately 110 lbs. per square inch. At maximum load the M.E.P. exceeded even 115 lbs. per square inch. This pressure is surprisingly high and in conjunction with high results for η_i and η_w points to excellent combustion. The compression pressure, 120 lbs. per square inch, exceeds by but little the inertia forces of the reciprocating parts whose weight was about 4.55 lbs. per square inch of piston.

6. Ganz & Co., Budapest. The attempt to obtain thermal efficiencies as high as possible by increasing the compression, while correct in theory, soon meets a limit in practice in that the increasing compression temperatures soon cause pre-ignition. To avoid this, various schemes of cooling the charge either by internal or external means have been tried. As early as 1888, Capitaine built a kerosene engine which took in a little water with the mixture, by means of which it was thought to be possible to prevent ignition, but he did not succeed in overcoming the practical difficulties involved. In 1894 Donat Bánki resorted to the same means of raising the possible compression in his engines, and five years later brought out an economically superior high-pressure engine, using water injection. This machine has since that time been built by Ganz & Co., mostly for operation with gasoline, although gas has lately also been tried (see below).

Considered from a purely thermal standpoint, the use of water in this manner affects the efficiency of internal combustion engines in an unfavorable sense, since it increases the specific heat of the mixture and decreases the temperature range.[1] The theoretical loss, however, seems to be overbalanced in the Bánki engine by the gain due to higher compression.

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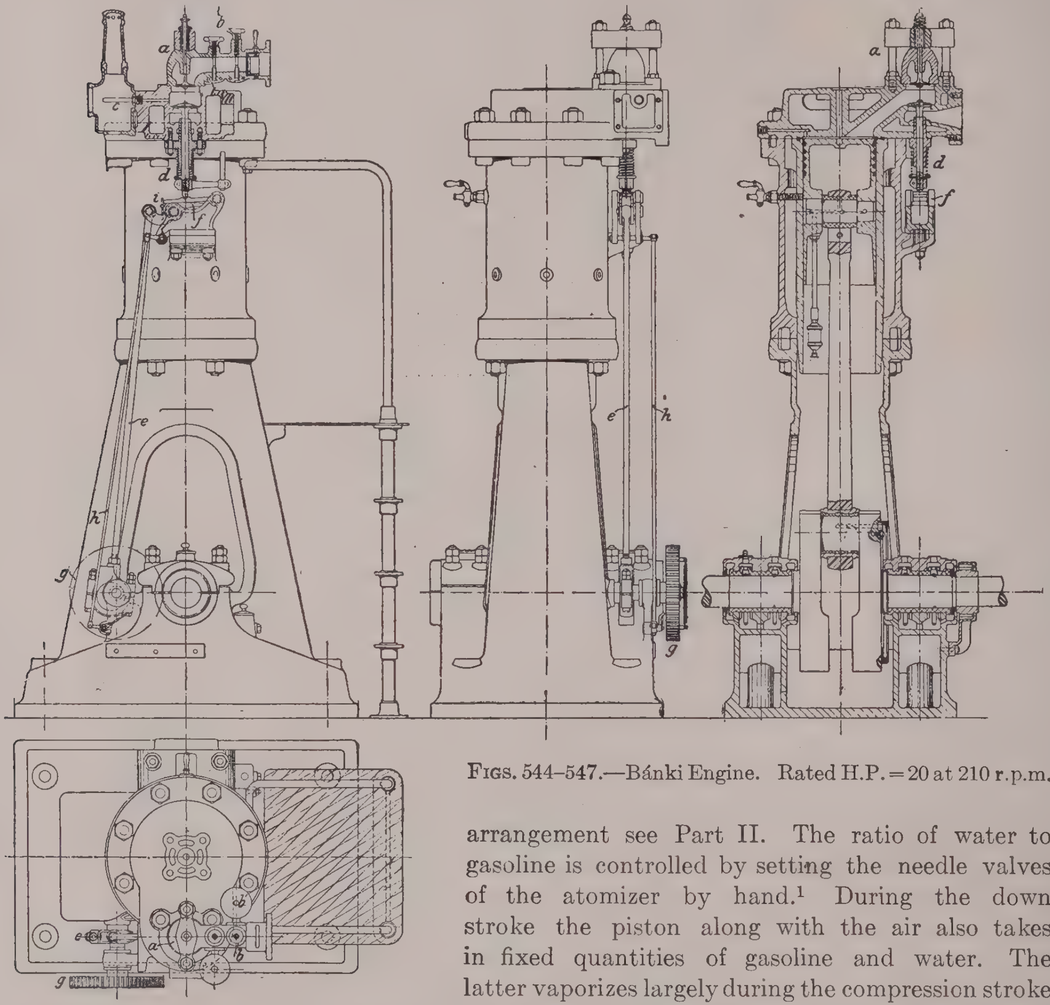
[1] Compare the investigation by Schreber in Dingler's Polyt. Journal, 1905, p. 33.

The general design of the Bánki engines is shown in Figs. 544–547. The only additional special arrangement made necessary by the method of operation is the double atomizer *bb*, placed ahead of the inlet valve *a*. For details of this atomizing arrangement see Part II. The ratio of water to gasoline is controlled by setting the needle valves of the atomizer by hand.[1] During the down stroke the piston along with the air also takes in fixed quantities of gasoline and water. The latter vaporizes largely during the compression stroke only and, in doing so, takes up so much heat that it becomes possible to compress to from 180 to 240 lbs. per square inch without pre-ignition. At the end of the up stroke the mixture is ignited by an open hot tube *c*, the pressure attained being from 500 to 550 lbs. per square inch. Expansion and exhaust follow as usual in 4-cycle engines.



FIGS. 544–547.—Bánki Engine. Rated H.P. = 20 at 210 r.p.m.

The exhaust valve *d* is operated by the eccentric gear *e* through the wiper cam *f*. The speed is controlled by holding open the exhaust valve. This is accomplished through the reach rod *h i*, the position of which is controlled by a shaft governor in

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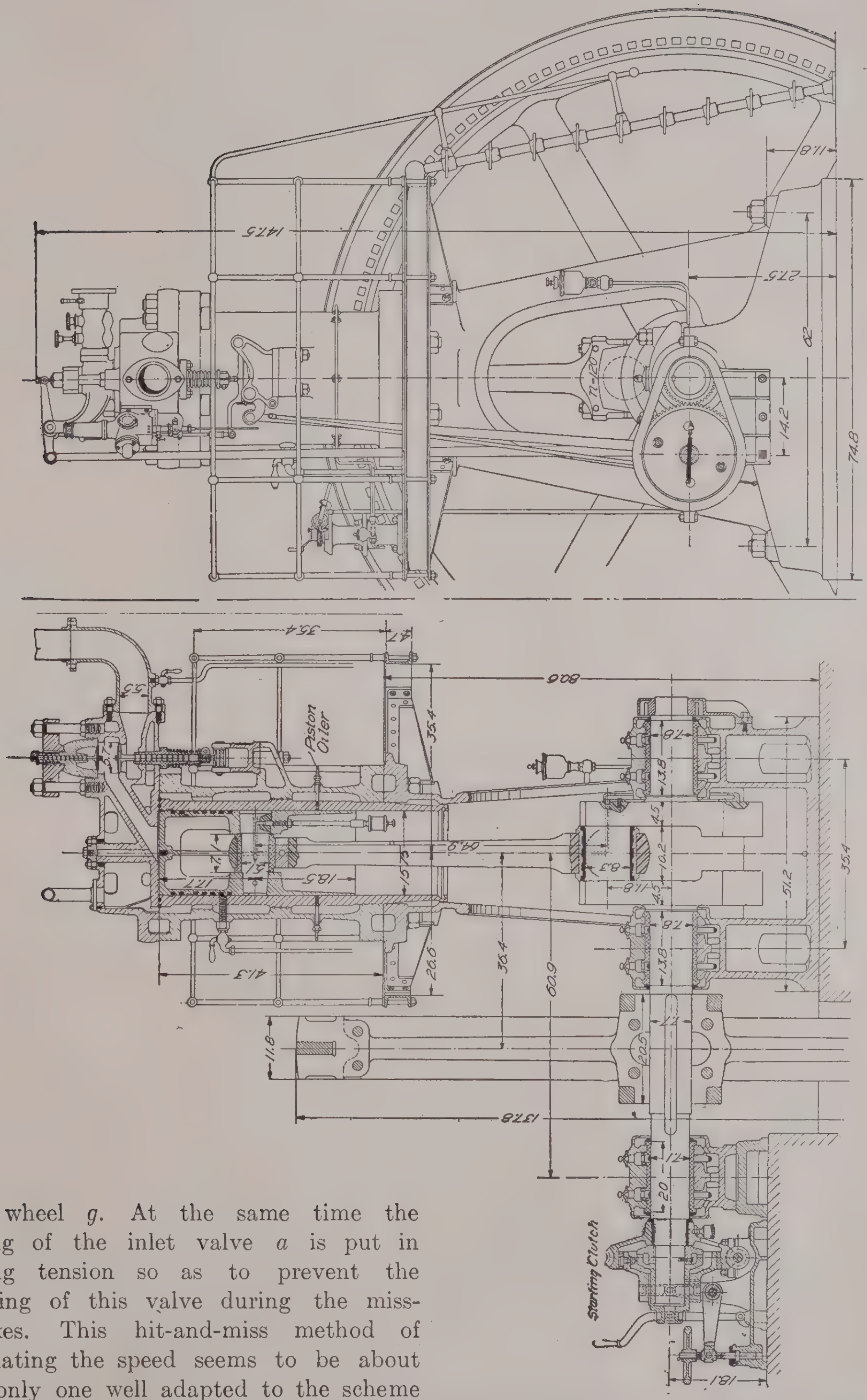
[1] German patent No. 77764.



Figs. 548 and 549.—Bánki Gasoline Engine. Rated H.P. = 50 at 120 r.p.m.

the wheel *g*. At the same time the spring of the inlet valve *a* is put in strong tension so as to prevent the opening of this valve during the miss-strokes. This hit-and-miss method of regulating the speed seems to be about the only one well adapted to the scheme of operation, since it would be difficult to satisfactorily proportion the quantities of gasoline and water to variations in load. But even in this case the author's own observations have shown that the cylinder temperature changes involved in the method of regulation may easily cause trouble.

Operating Results. (*a*) Tests made by Prof. E. Meyer, 1899, on a 20-H.P. engine, $D=9.85''$, $S=15.70''$, compression space, .0785 cu.ft. The gasoline used had a specific gravity of .70 and a lower heating value of 18520 B.T.U. per pound.[1]

TABLE 70

Test No.	1	2	3	4	5
R.p.m.	210.9	211.5	212.4	214.4	216.2
B.H.P.	24.9	19.3	13.1	6.67	0
Gasoline per B.H.P. hour . . . lbs.	.541	.590	.634	.850	3.26 per hr.
Injection water per hour . . . lbs.	71.7	41.6	34.8	18.15	11.22
Ratio $\frac{\text{water injected}}{\text{gasoline}}$	5.34	3.93	4.20	3.21	3.45
Cooling water per B.H.P. hour . . . lbs.	40.0	56.3	72.8	86.2	. . . .
Per cent heat carried off in cooling water	22.4	26.0	31.8	28.8	. . . .
Economic efficiency . . . %	25.3	23.2	21.6	16.1	. . . .

The mean effective pressure, assuming $\eta_m=.85$, was computed to be 108 lbs. per square inch, which is rather high for an explosion engine. The indicator cards show explosion pressures up to 540 lbs., with compressions at 233 lbs. per square inch. In the consumption figures above quoted, the gasoline required by the heating lamp for the hot tube, which amounted to .418 lbs. per hour, is not taken into account.

(*b*) Tests by Professor Jonas, with the assistance of Mr. Taborsky, made on the same engine near the end of 1899. The gasoline had a specific gravity of .7298 and a lower heating value of 18 322 B.T.U. per pound.

TABLE 71

Test No.	1	2	3	4	5
R.p.m.	209.13	209.67	209.83	210.50	210.7
B.H.P.	26.00	20.40	14.85	8.10	0
Gasoline per B.H.P. hour . . . lbs.	.493	.523	.582	.727	3.4 per hr.
Injection water per hour . . . lbs.	62.20	35.3	24.40	13.72	10.20
Ratio $\frac{\text{water injected}}{\text{gasoline}}$	4.84	3.30	2.82	2.33	3.00
Cooling water per B.H.P. hour	29.80	45.50	37.70	51.80	. . . .
Heat lost in cooling water, per cent of total	21.7	26.7	23.6	27.6	. . . .
Temperature exhaust gas . . . °F.	383	383	366	340	232
Economic efficiency . . . %	28.0	26.4	23.8	19.0	. . . .

The fuel consumption in this case is even a little better than it was in Professor Meyer's tests. The effect of the water injected is quite strongly manifest in the low temperature of the exhaust gases. The explosion pressure varied between 555 and 654 lbs. per square inch, depending upon the load, with compression at 234 lbs. per square inch.

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[1] Z. d. V. D. I., 1900, p. 1062.

As above mentioned, attempts were made to operate engines with gas on the same principle. The development apparently extended over several years and has shown results as indicated below. It is, however, still a question whether the Bánki gas engine has attained marketable status. The firm itself is as yet a licensee of one of the builders of German gas engines.

(*c*) The author has received the following tests of a 16-H.P. gas engine from Professor Bánki himself:

Load	Full	Half	No Load
R.p.m.	255	256	258
Explosions per minute	122.6	76	22
B.H.P.	16.8	8.61	0
Injection water per hour ... lbs.	22.75	16.55	1.87
Gas per B.H.P. hour ... cu.ft.	13.89	16.22	51.2 per hr.
Injection water per B.H.P. hour ... lbs.	1.355	1.92	....
Economic efficiency ... %	32.43	27.79	....

The last line seems to indicate that the heating value of the gas was in round numbers 560 B.T.U. per cubic foot.

(*d*) Professor Schimanek, who made the above series of tests, stated in addition that the above figures had up to now been obtained only in two cases and that, probably on account of enriching the mixture by excessive use of cylinder oil (excess approximately 60%), the figures show results for gas consumption that are too low. In five other tests, made with the piston only sparingly lubricated, Schimanek found on the average the following results:

Load, B.H.P.	16.75	8.58	0
Gas used per B.H.P. hour ... cu.ft.	15.32	17.80	55.0 per hr.

referred to gas of 560 B.T.U. per cubic foot at 32° F. and 760 mm. Hg.

The diagram, Fig. 550, with compression at 228 lbs., shows an explosion pressure exceeding 780 lbs. per square inch. This value, however, seems much too high and apparently represents an explosion accidentally very heavy. It is also quite probable that of this pressure at least 70 lbs. may be ascribed to inertia in the indicator. A good insight into the real pressure variations during the cycle is given by Fig. 551, which represents a distorted diagram obtained on a large scale by means of an optical indicator (free from inertia). The dead center positions T_{p_i} at the end of compression and T_{p_a} at the end of expansion have of course been indicated by hand afterwards.

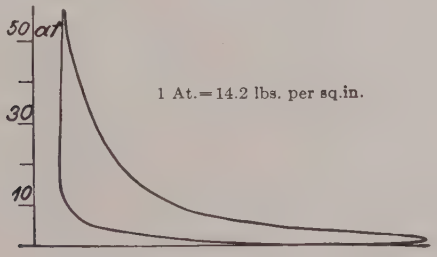


FIG. 550.

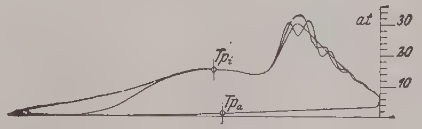
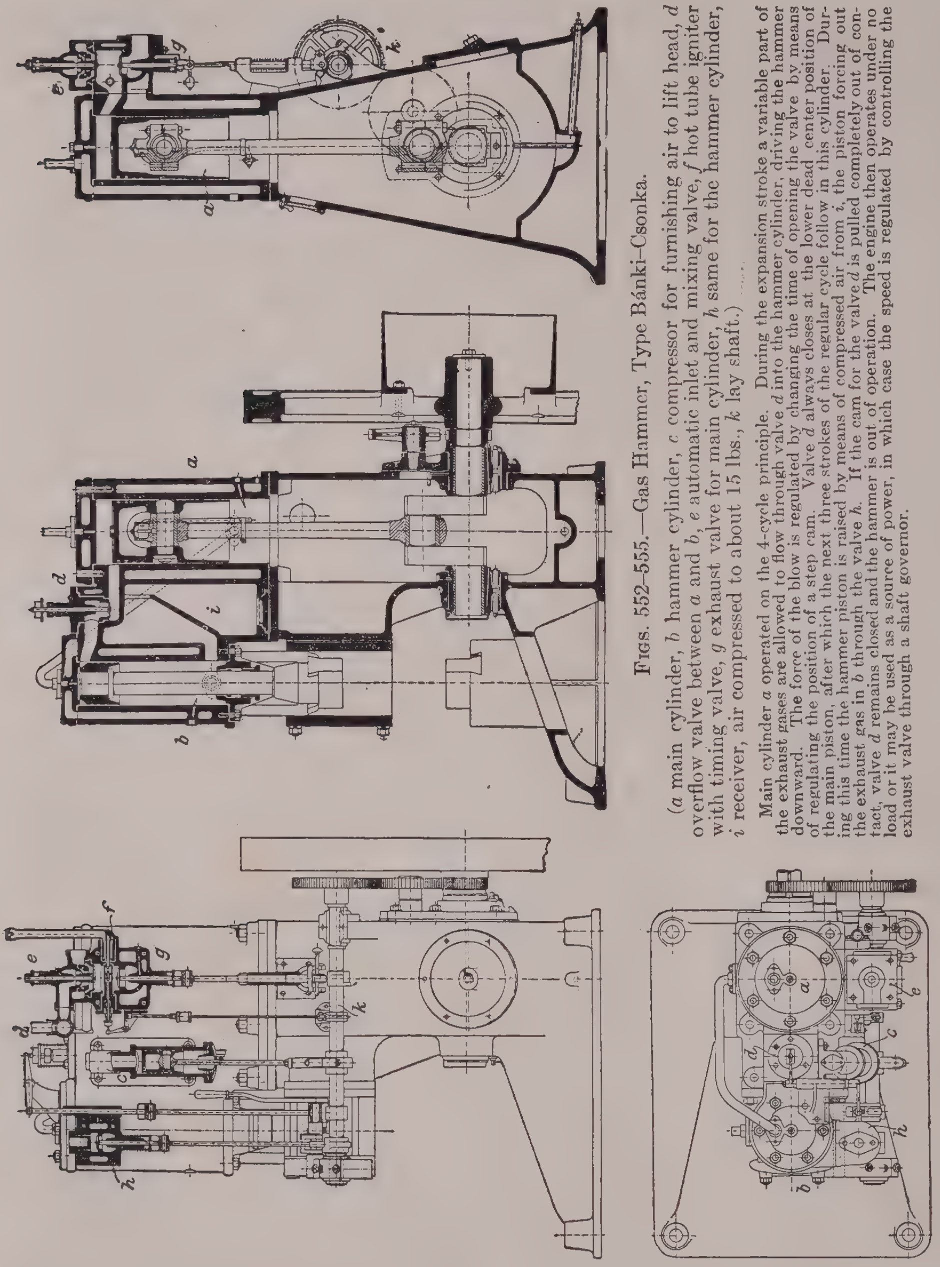


FIG. 551.



FIGS. 552–555.—Gas Hammer, Type Bánki–Csonka.

(*a* main cylinder, *b* hammer cylinder, *c* compressor for furnishing air to lift head, *d* overflow valve between *a* and *b*, *e* automatic inlet and mixing valve, *f* hot tube igniter with timing valve, *g* exhaust valve for main cylinder, *h* same for the hammer cylinder, *i* receiver, air compressed to about 15 lbs., *k* lay shaft.)

Main cylinder *a* operated on the 4-cycle principle. During the expansion stroke a variable part of the exhaust gases are allowed to flow through valve *d* into the hammer cylinder, driving the hammer downward. The force of the blow is regulated by changing the time of opening the valve by means of regulating the position of a step cam. Valve *d* always closes at the lower dead center position of the main piston, after which the next three strokes of the regular cycle follow in this cylinder. During this time the hammer piston is raised by means of compressed air from *i*, the piston forcing out the exhaust gas in *b* through the valve *h*. If the cam for the valve *d* is pulled completely out of contact, valve *d* remains closed and the hammer is out of operation. The engine then operates under no load or it may be used as a source of power, in which case the speed is regulated by controlling the exhaust valve through a shaft governor.

According to this card, the explosion pressure is only about 470 lbs. when the compression is 220 lbs. per square inch. It is also clear from the distorted diagram that the explosion is by no means as rapid as is indicated in the ordinary diagram, Fig. 550. Of course the bundle of diagrams in Fig. 551 shows nothing but explosions that are considerably delayed, and it may therefore be assumed that the normal explosion would show a more rapid rise of the line. Shortly before the end of the compression, T_{p_i}, there is a marked drop in the line which is apparently due to loss of mixture past the piston.

Since 1892 the firm of Ganz & Co. have also built what may be considered the oldest gas-hammer. It seems quite natural to apply the explosive force of gas-air mixtures to the operation of hammers, pile drivers, etc., and the scheme has often been tried,[1] but only the Ganz & Co. hammer seems to have been able to maintain itself. The latter, constructed according to the patents held by Bánki and Csonka, is made only in one size, Fig. 552–555, and has the following dimensions:

Diameter of main cylinder	8.67″
Diameter of hammer cylinder	5.90″
Stroke of the main piston	12.60″
Stroke of the hammer piston	13.75″
Diameter of hammer rod	2.75″
Weight of hammer piston, rod and hammer	110 lbs.
R.p.m. of crank-shaft	220
Number of hammer blows per minute	110
Variable kinetic energy of one stroke,	from 0 to 2880 ft.-lbs.

IV. American Gas Engines

1. The Westinghouse Machine Co., East Pittsburg, Pa. This company has been building vertical gas engines since 1896, and since about 1903 has also been building horizontal engines. All their product has been of the 4-cycle throttle governing type. Their vertical engines have all been two- or three-cylinder machines with box type crank case enclosing the working parts and providing splash lubrication. Several more or less distinct types have been brought out.

Fig. 556 shows the earliest type, while Figs. 557–560 show a later form. The engines are designed to operate on natural, illuminating, or producer gas.

The two-cylinder engines have been built in sizes from 10 to 85 B.H.P. and the three-cylinder engines from 40 to 650 B.H.P. The horizontal engines are made in large sizes only, and since their advent this company no longer builds vertical engines larger than 360 B.H.P.

The principal features of the *vertical engine* may best be studied in Figs. 558 and 559. The exhaust valves are located in the cylinder casting at the rear of the engine and discharge into a common manifold. They are operated by a cam shaft inside the crank case.

The inlet valves are located in the front part of the cylinder cover and are operated by a cam shaft mounted above the mixture inlet manifold.

An auxiliary igniter cam shaft parallel to the inlet cam shaft is geared to it and serves to vary the time of ignition while running to suit variations in the quality of the gas.

The centrifugal governor controls the lift of the balanced valve. The proportion of gas and air admitted to this valve is regulated by the hand adjustment of cocks (not shown) in the gas and air lines. These serve to adjust the mixture when the quality of the gas changes.

---

[1] See, for instance, Z. d. V. D. I., 1894, p. 582; Richard, Nouveaux Moteurs à Gas III, p. 816; American Machinist, 1884, Dec. 24.

Ignition is of the hammer break type, current for the spark coil being taken from either a small 100-volt dynamo through resistance lamps or from a primary or storage battery.

The engine is started by compressed air introduced by special valves into one of the cylinders, which is for the time arranged to act as an air motor. The smaller engines are provided with a pulley to charge the compressed air tanks after the engine has started. Large engines are usually accompanied by a small independently driven air compressor.

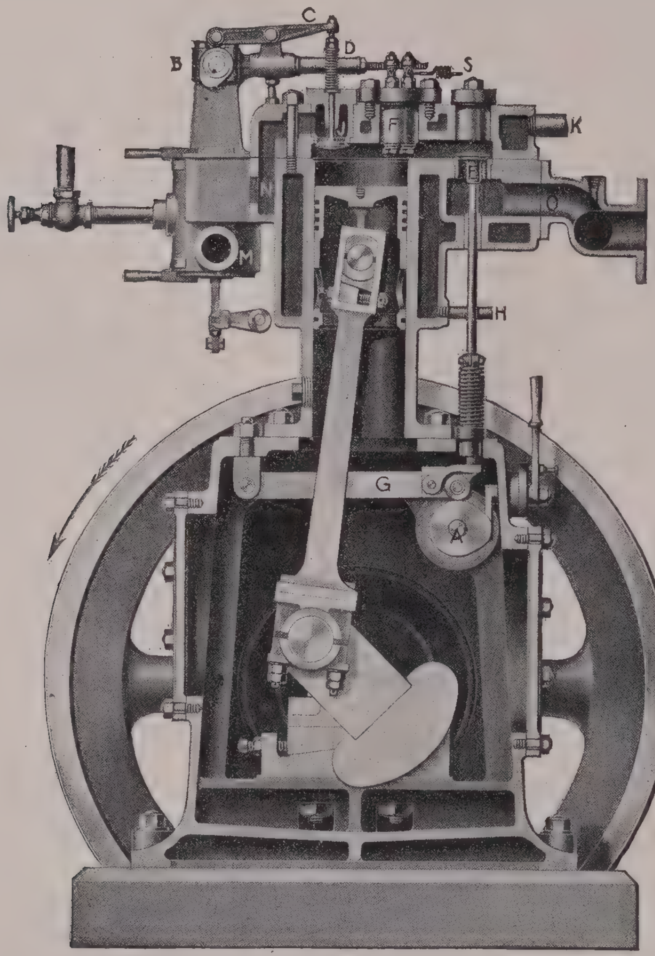


Fig. 556.—Westinghouse Gas Engine.

The following description of the Westinghouse horizontal engines is taken from two sources, the *Report of the Committee on Gas Engines* of the National Electric Light Association, and *Power*, April 28, 1908. The engine is of the double-acting tandem or twin-tandem type. In the latter case the cranks are 90° apart.

Fig. 561 shows one of the latest tandem installations, that of the new plant of the Union Switch and Signal Co. at Pittsburg. This is the first direct connected 60-cycle plant to go into service in this country with single-crank double-acting engines.

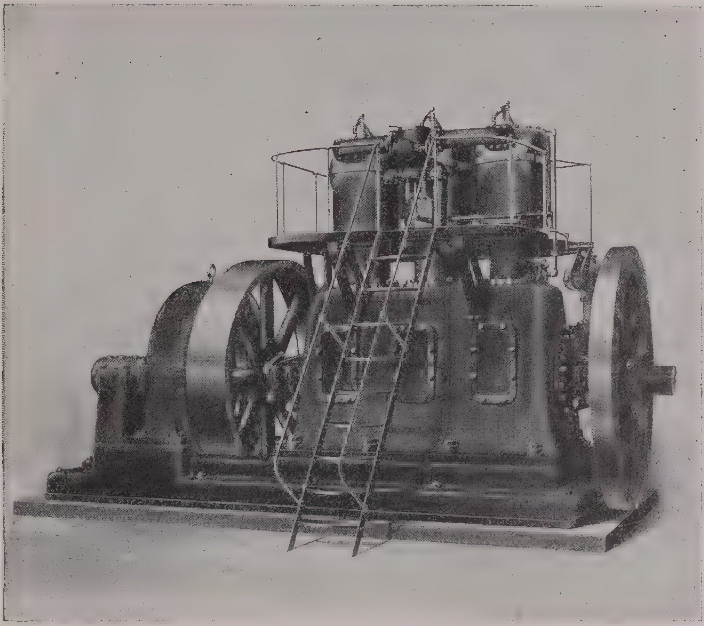


FIG. 557.—Front View, Westinghouse Vertical Gas Engine.

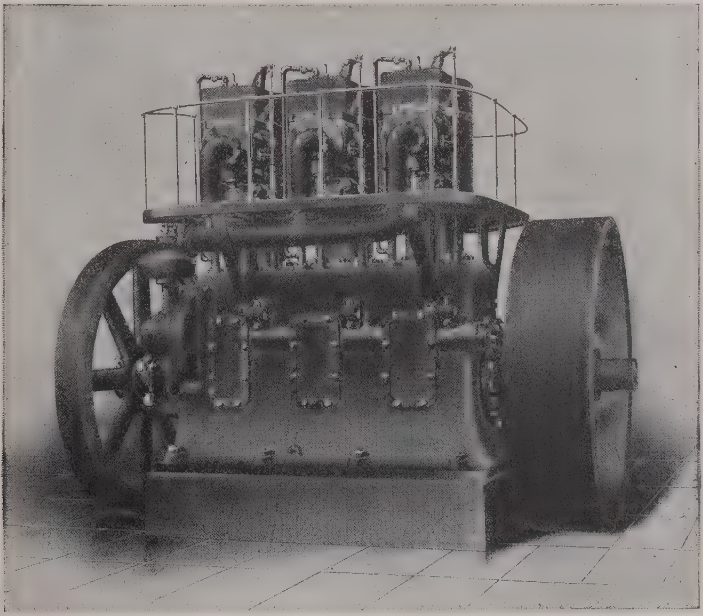


FIG. 558.—Back View, Westinghouse Vertical Gas Engine.

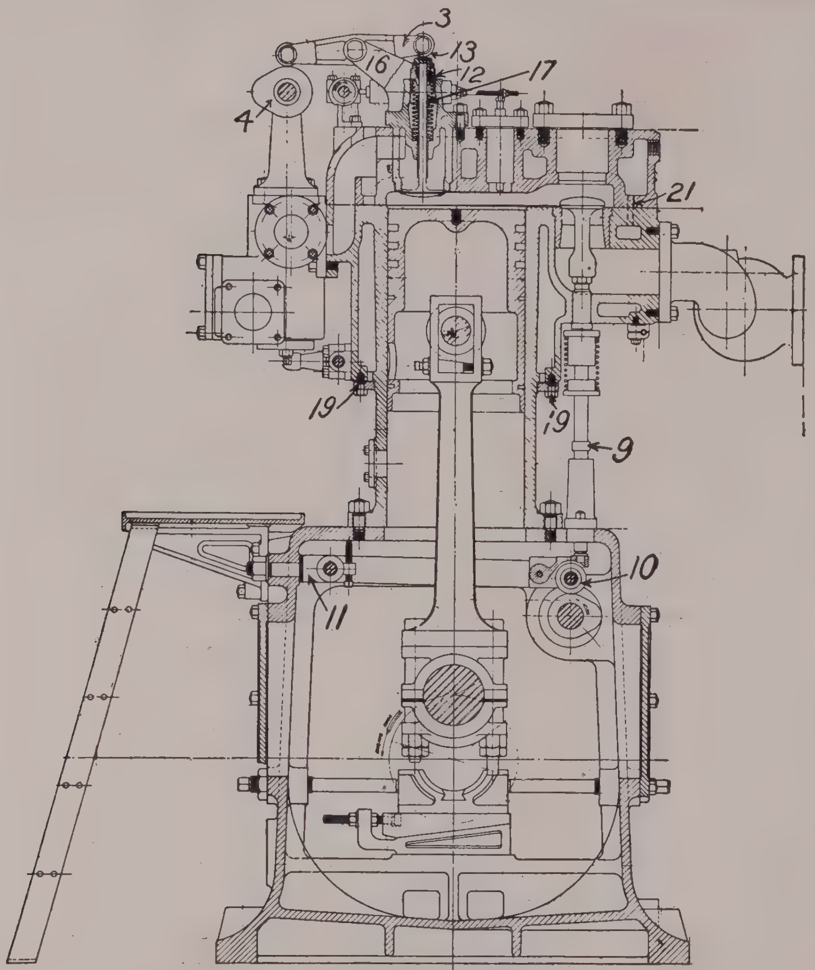


FIG. 559.—Cross-section, Westinghouse Vertical Engine.

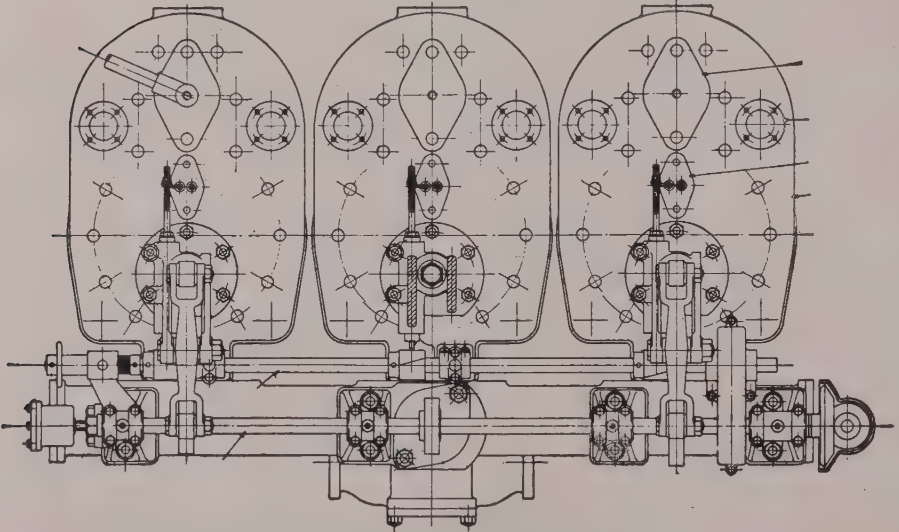


FIG. 560 —Top View, Westinghouse Vertical Engine.

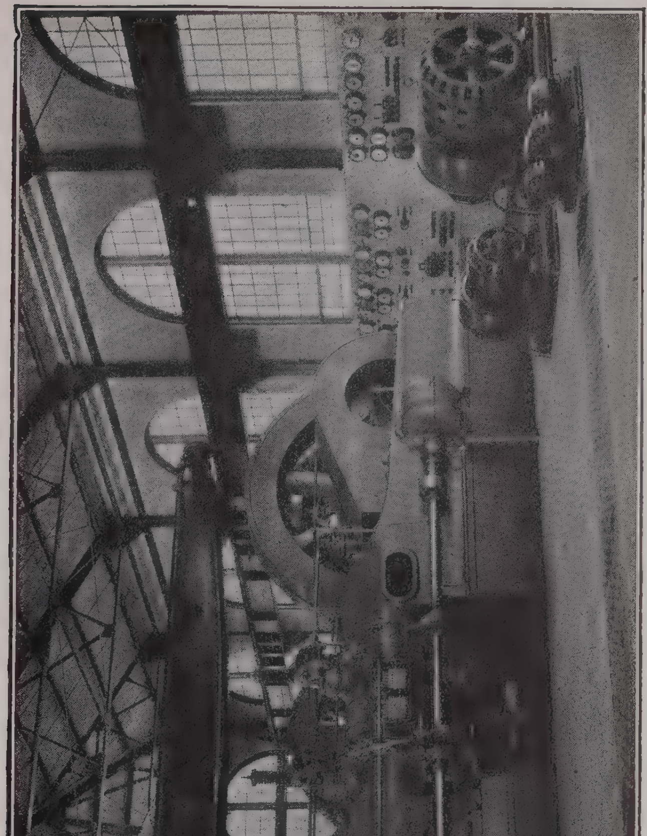


FIG. 561.—Westinghouse Horizontal Gas Engine, Plant of the Union Switch and Signal Co., Pittsburg, Pa.

Figs. 562–567 will serve to show the general features of design. In the main these are as follows:

The *main frame* is of heavy girder box construction, the girder surrounding the crank pit to stiffen the frame against the transverse stresses due to the use of the side crank.

The *center* and *rear housings,* see Fig. 566, are approximately cylindrical constructions cut away at the top to give access to the center and tailrod slippers. To reinforce the center housing and to distribute the strains more evenly, removable steel struts are placed above the center line as shown.

The *cylinders,* see Fig. 562, are supported at the ends only, giving complete access to the exhaust valves and piping below the cylinders. In the smaller sizes of engine the cylinders are symmetrical one-piece castings, having a divided jacket wall to avoid skrinkage stresses. The opening in the jacket wall is closed by split jacket bands with joints of flexible packing to admit of independent expansion of cylinder and jacket walls. The cylinders of the larger sizes are cast in halves to obtain sound metal at the center of the cylinder, and these halves are then joined by means of a ground joint and shrink links. See Fig. 567.

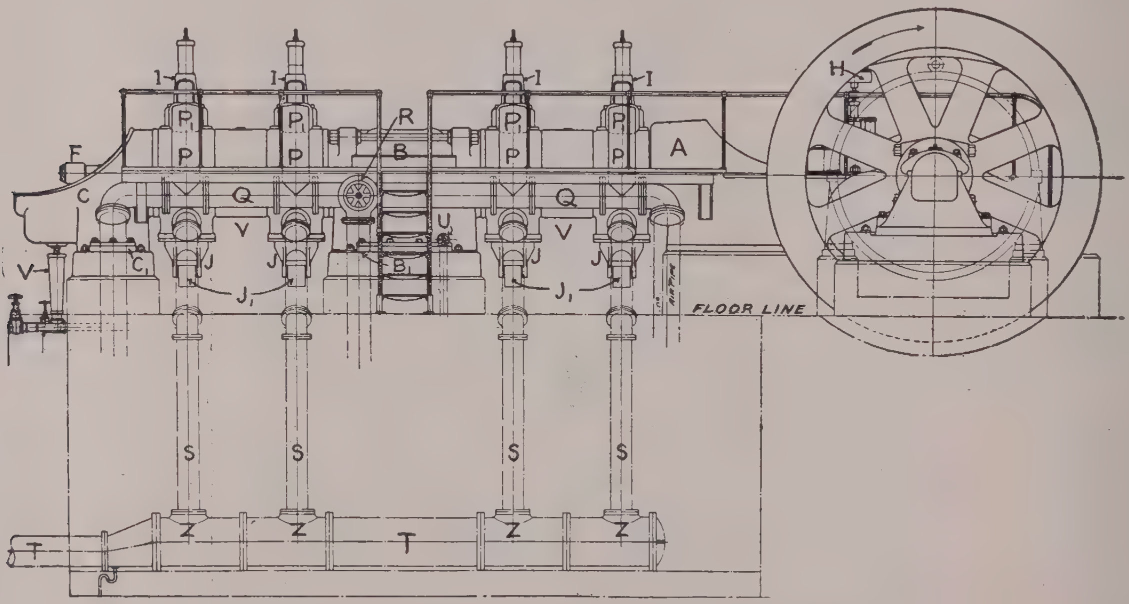


FIG. 562.—General Elevation, Westinghouse Horizontal Engine.

Only the main frame is rigidly anchored to its foundation. Both the center and rear housings are free to slide on their base plates, hence permitting the unrestrained expansion of the entire construction.

Quick access to the cylinders may be had through the inlet valve openings; no heads need to be removed.

The *pistons* are cast in one piece without chaplets. The piston is forced on the rod with a tapering fit and held in place by a rotating nut which is turned off flush with the piston face after setting up.

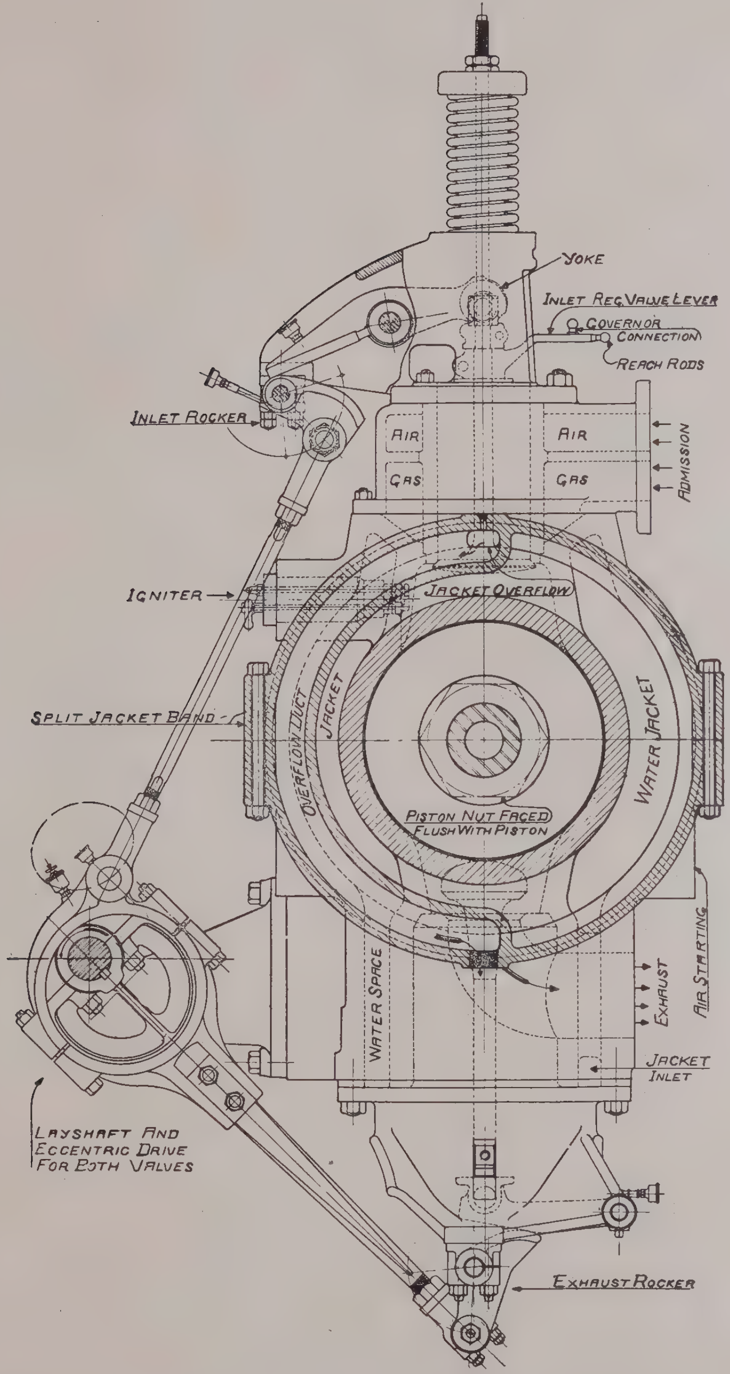


Fig. 563.—Cross-section through Cylinder, Westinghouse Horizontal Engine.

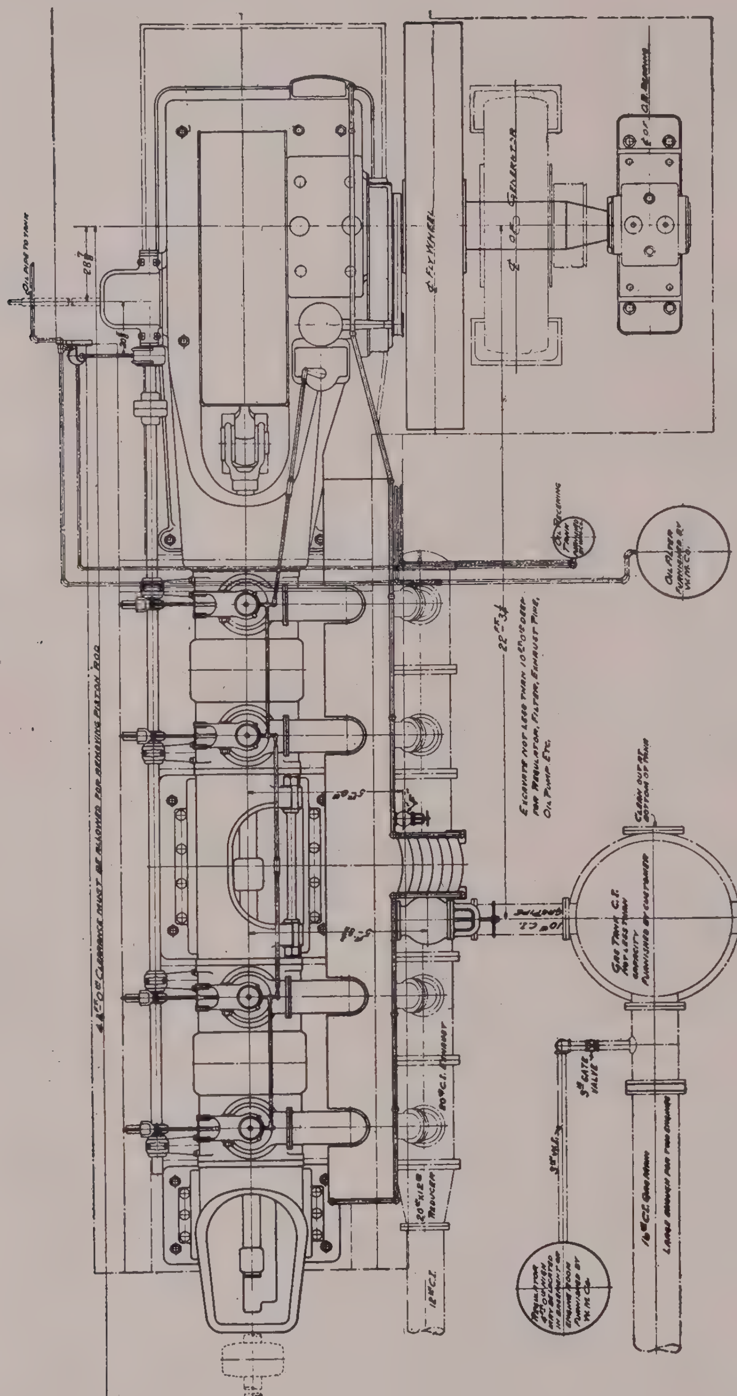


FIG. 564.—Plan View, Westinghouse Horizontal Engine.

The piston rod is in two parts, so that the latter may be removed through the front and rear housings.

The above mentioned report of the National Electric Light Association describes the *valves* and *valve gear* as follows:

Valves. Instead of a single governor valve controlling all inlets from a single point, the governing is accomplished directly at each inlet (Fig. 563), the inlet valve performing the function of both mixing and governing. Thus gas and air are mixed only at the point of entry to the cylinder, and in exact quantity required by the load—a condition essential for sensitive governing. The inlet valve (Fig. 563) has two distinct motions—a fixed *vertical motion* due to the eccentric, and *rotation* of an internal sleeve controlled only by the governor and provided with ports registering with corresponding ports in the surrounding valve cage. The essential function of this rotating sleeve is that of a purely throttling governor which opens or closes the respective ports according to the load. Hence the mixture itself remains *unchanged* at any load. Furthermore, the quantity of mixture that can be affected by a back-fire is a comparatively small volume contained within the inlet valve sleeve.

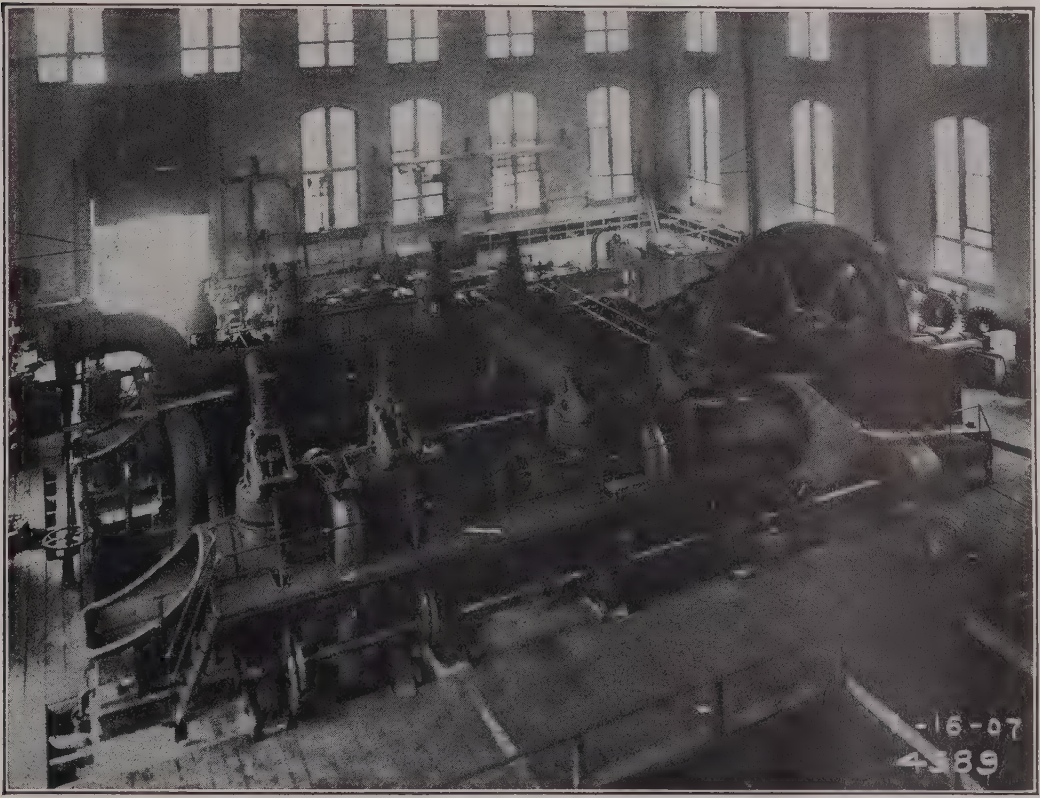


Fig. 565.—View showing Valve Gear, Westinghouse Horizontal Engine.

Valve Gear. An excellent feature of this gear is that only one eccentric is employed to operate both inlet and exhaust valves, instead of independent cams or eccentrics (Fig. 563). This eccentric motion is transmitted by pull rods to a rolling-cam motion similar to that used in marine work. Inlet and exhaust are, of course, reversed to accomplish the valve openings at proper points in the cycle. This motion gives the very desirable maximum power at opening, followed by rapidly increasing lift as the valve opens, without bringing heavy thrusts upon the eccentric.

The lay shaft carrying these eccentrics is driven from the main shaft by spur and bevel gear in place of the usual spiral gearing. This is designed to avoid the back-lash which eventually develops with spiral gears, due to the variable torque encountered. With fixed eccentrics and no lost motion, it is evident that no opportunity exists for change of adjustment of valves except from wear of the rolling cams, which is obviated by the use of generous contact areas well lubricated. Erratic wear of the gear drive is prevented by the use of the "hunting tooth," *i.e.*, an odd number in each reduction.

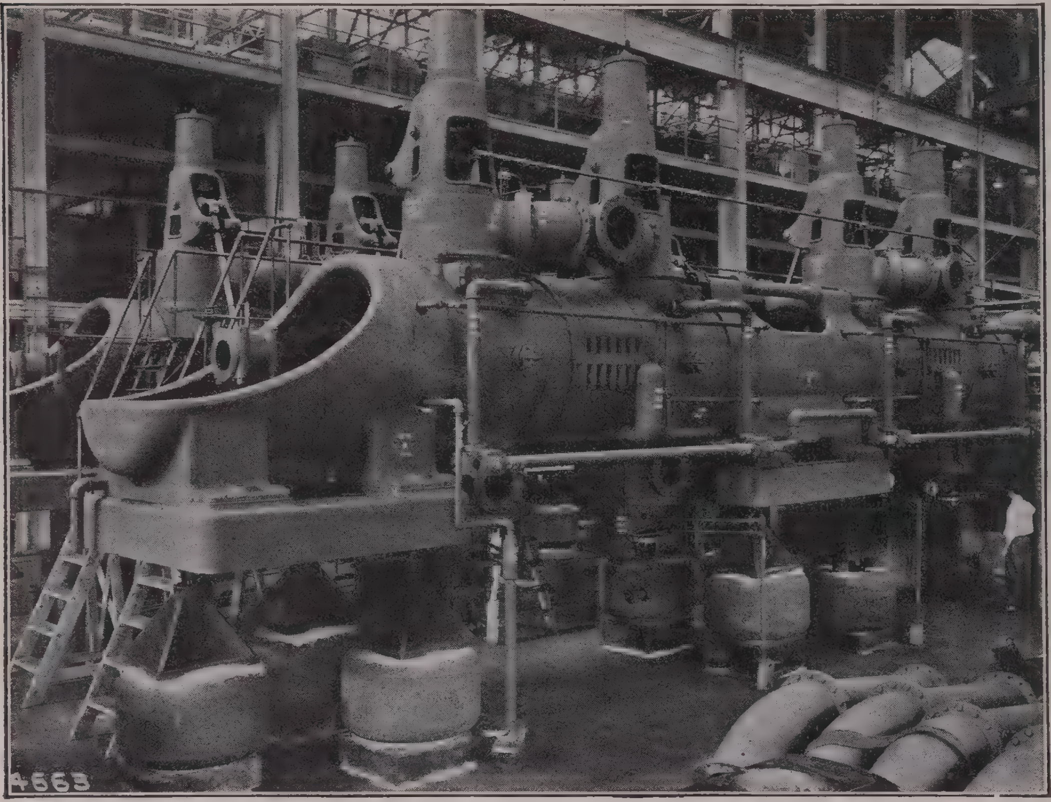


FIG. 566.—View showing Center and Rear Housing, Westinghouse Horizontal Engine.

Exhaust Valves. Hollow water-cooled valves of either cast iron or steel are used in all sizes, or a combination valve with cast-iron head and steel stem. Inside the bore a small tube rises to the head of the valve, constituting an overflow pipe. Cold water ascends through the outer annulus. The internal pipe not only prevents air pockets being formed but also insures that the valve never becomes dry. Stems are lubricated about the middle of the bushing. To remove the valves it is only necessary to drop out the entire cage with the assistance of a rope swing.

The *governor* is of the Hartung or Jahns type, and is directly driven from the main shaft. It does not itself act upon the system of reach rods used to operate the regulating valve at each inlet, but simply operates a small balanced pilot valve which admits oil under pressure to one or the other end of a cylinder containing a piston, the rod of which controls the movements of the reach rods. By means of an oil relay system of this type the governor itself is relieved of nearly all work.

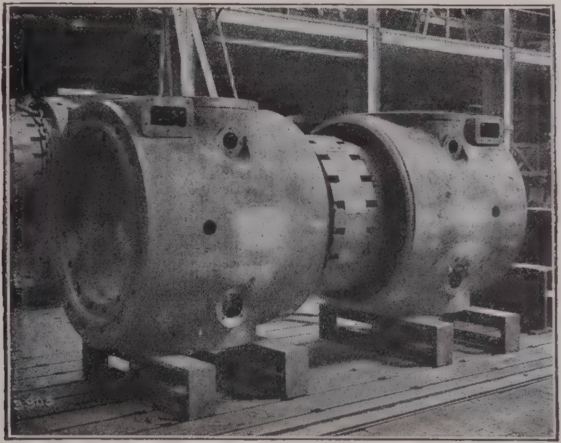


Fig. 597.—Cylinder Construction, large Westinghouse Horizontal Engine.

The *ignition* system of this engine deserves special mention. Make-and-break igniters are used, in all cases at least two, and in the larger engines three, in each combustion chamber. Both sides of each igniter are thoroughly insulated, making a double ground necessary to prevent successful operation.

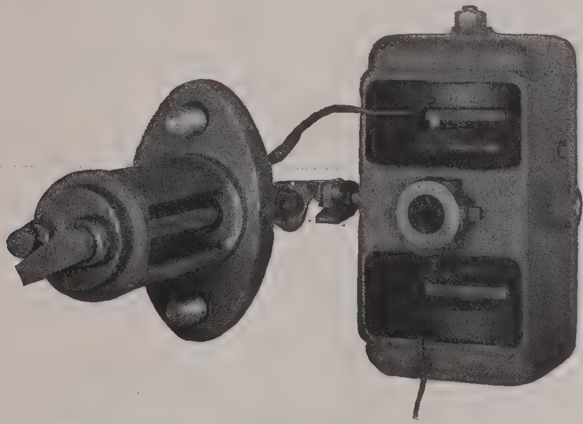


Fig. 568.—Igniter Block, Westinghouse Horizontal Engine.

Two types of make-and-break igniters are used. One of these is the well known mechanical knock-off type and needs no further description; the other may be termed electro-mechanical. The igniter block used in the latter system is shown in Figs. 568 and 569, while Fig. 570 shows the wiring diagram. The electro-magnet shown in Fig. 568 has a working armature which strikes the movable electrode of the igniter plug proper when the circuit is established. The coil is in series with the igniter electrodes and serves as an induction coil. The circuit is made and broken by the commutator or timer shown in Fig. 570. This acts exactly like the well-known commutator used in the primary circuit of jump-spark systems, and the point of ignition may be changed in the usual way by revolving the case carrying the various circuit terminals. The electro-magnet acts as a tell-tale to indicate whether the igniter is working. The system also includes a safety stop. (See Fig. 570.) Should the speed of the engine become excessive, a dog is thrown out from the face of the fly-wheel by centrifugal force. This strikes the trigger of the trip device, breaks the circuit and puts all of the ignition out of operation.

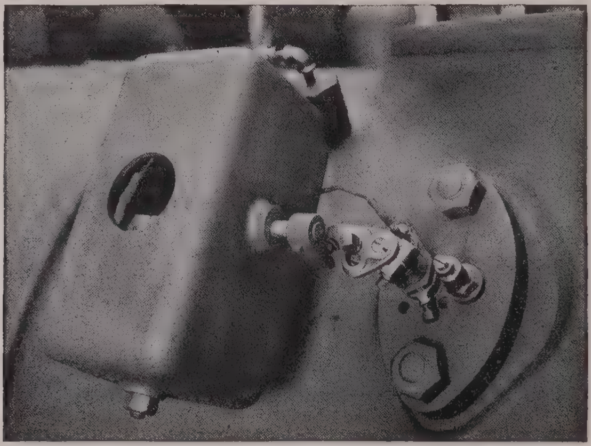


Fig. 569.—Igniter Block; Westinghouse Horizontal Engine.

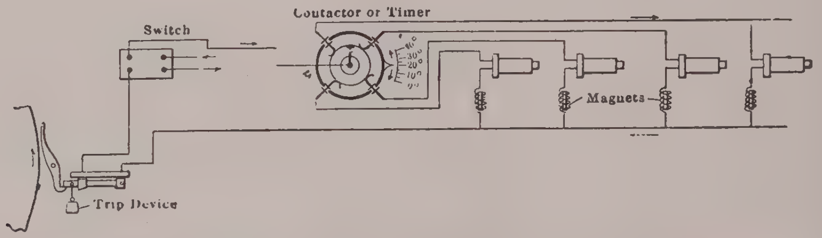


Fig. 570.—Ignition System, Westinghouse Vertical Engine.

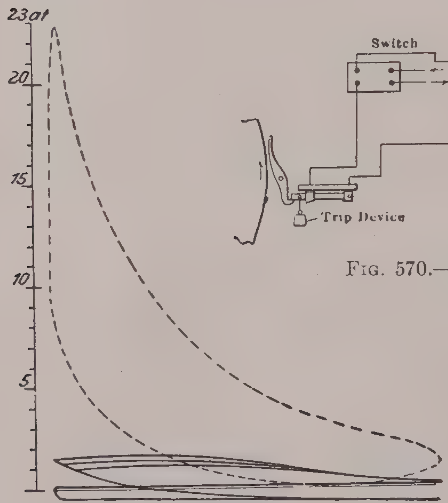


Fig. 571.—Diagrams from Westinghouse Vertical Engine.

Force feed lubrication is used for the cylinders, packing glands and exhaust valve stems. The oil supply to the cylinder is so timed as so reach the piston at the end of the expansion stroke. There are thus in all cases two low temperature strokes available for good distribution. The exact time of supplying the oil may be controlled by adjusting the eccentrics driving the oil pumps. For the other engine parts the usual gravity feed system is employed, but so arranged that only a single valve needs to be opened and closed.

Operating Results.

VERTICAL ENGINE. (*a*) Test made by C. H. Robertson, 1900. Three-cylinder engine rated at 125 B.H.P., cylinder diameter = 13″, stroke = 14″. Heating value of the natural gas used as fuel was on the average 970 B.T.U. per cubic foot. Duration of each trial about one hour.

The ratio of gas to air in all six trials was 1 to 13.7. The no-load diagram, Fig. 571, clearly shows the throttling of the charge and marked after-burning. The diagram in broken line is that obtained at maximum load, Test No. 5.

TABLE 72

Test No.	1	2	3	4	5	6
Load	$\frac{1}{4}$	$\frac{1}{2}$	$\frac{3}{4}$	Rated.	Maximum.	No Load.
R.p.m.	264	264	260	258	256.88	265.5
Comp. pressure ..... in lbs.	40.5	60.3	70.3	110.0	130.0	25.5
Explosion pressure.. middle per	61.7	99.0	159.0	238.0	300.0	32.6
Mean eff. pressure .. cyl. sq.in.	27.7	46.8	60.5	80.0	90.0	8.4
I.H.P.	53.69	85.55	114.69	139.73	164.22	18.57
B.H.P.	30.81	64.37	90.25	112.86	143.17	0
Mechanical efficiency ....%	57.38	75.24	78.69	87.93	87.0	0
Gas per B.H.P. hour ....cu.ft.	24.3	14.65	12.75	11.80	11.70	470 per hr.
Thermal efficiency ....%	15.6	23.8	26.3	25.5	26.1	10.2
Economic efficiency ....%	10.76	17.96	20.53	22.37	22.56	......
Exhaust temperature ....°F.	1510	1624	1722	1832	......	......
Heat lost in cooling water ....%	53.6	......	37.0	32.11	29.6	57.6
Heat lost in exhaust and radiation %	30.8	......	36.7	42.4	44.3	32.2

(*b*) Tests of Miller & Gladden, 1900. Three-cylinder engine, rated at 550 B.H.P., cylinder diameter 25″, stroke 30″. Pistons and exhaust valves water-cooled. Heating value of Pittsburg natural gas on the average 1000 B.T.U. per cubic foot. Duration of tests 1 to 3, each 5 hours, Test No. 4, 2½ hours.

TABLE 73

Test No.	1	2	3	4
Load	$\frac{1}{3}$	$\frac{2}{3}$	Normal.	Maximum.
R.p.m.	151.7	150.0	149.2	146.7
Mean effective pressure, pounds per square inch.	31.0	53.8	75.3	82.7
I.H.P.	262.5	449.3	621.4	676.7
B.H.P.	207.7	383.8	553.1	605.6
Mechanical efficiency	79.15	84.72	88.99	89.71
Gas per B.H.P. hour ....cu.ft.	15.7	11.5	10.45	10.05
Thermal efficiency ....%	20.51	25.80	27.35	28.38
Economic efficiency ....%	16.23	22.02	24.30	25.46
Cooling water per B.H.P. hour ....gals.	5.62	4.13	3.66	3.54
Cooling water to cylinder and exhaust .... % total	70.35	72.78	72.22	72.32
Cooling water to piston	29.66	27.22	27.78	27.67
Mean temperature rise of cooling water .... °F.	88.2	84.8	79.2	79.2
Heat lost in cooling water ....%	38.69	34.39	33.34	33.60
Heat lost in exhaust and radiation ....%	40.80	39.83	39.33	38.01

The figures reported in these tables deserve special attention, particularly the high mean effective pressure of 90 lbs. per square inch and the excellent mechanical efficiency of nearly 90%. The thermal efficiency also is very high. The effect of the short cylinder jacket manifests itself in two directions, in a favorable sense as far as the low consumption of cooling water is concerned, and unfavorably in that the exhaust temperatures are high.

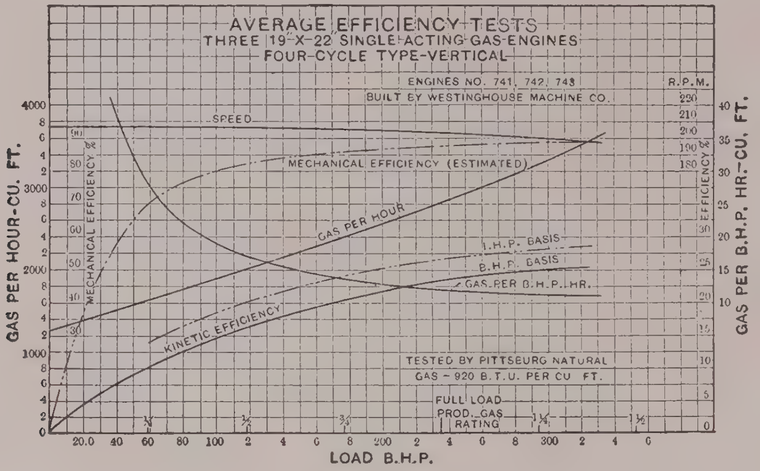


Fig. 572.

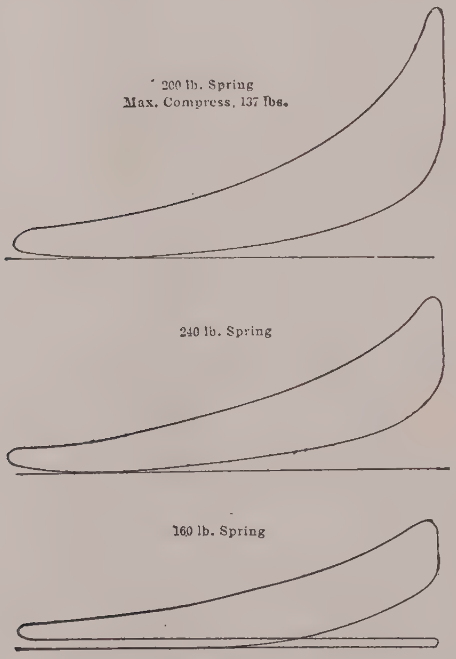


Fig. 573.

(*c*) Tests reported by J. R. Bibbins to the American Society of Mechanical Engineers, December, 1906. These tests were made on Westinghouse three-cylinder vertical engines before and after they were installed in the plant of the Gould Coupler Co. at Depew, N. Y. The fuel used is producer gas made in Loomis-Pettibone pressure producers operating on bituminous coal. The load on the plant, although to some extent equalized by overlapping demands from various sources, fluctuates widely, 80% above or below the general average, and the results obtained in the plant should on that account be of special interest.

The plant capacity is 450 K.W. Each of the three engines is rated at 235 B.H.P. on producer gas, capable of developing 260 B.H.P. as a maximum. The cylinder diameter is 19″, the stroke 22″, while the normal speed is 200 r.p.m.

The engines were tested in the shop of the company at East Pittsburg on natural gas. The results of these tests are given in Table 74, and graphically in

Fig. 572. The term "kinetic efficiency" corresponds to what is called thermal efficiency throughout this book, "brake kinetic efficiency" corresponding to economic efficiency. Fig. 573 shows some typical indicator cards obtained in these tests.

TABLE 74

ENGINE EFFICIENCY

AVERAGE OF TESTS

(Three 19×22″ single-acting 4-cycle gas engines. Engines Nos. 741, 742, 743.)

Load.	1⅓	Full	½	Remarks.
Brake H.P.	325.3	239	121.3	By Prony brake
Speed r.p.m.	198	203	206.3	By counter
Gas per hour	3547	2840	1985	Corr. to 60° F., 30″ Hg.
Gas per B.H.P. hour	10.9	11.87	16.36	Corr. gas—no load—1250 ft. per hr
Heat value gas *	920	920	920	Effect B.T.U. per cu.ft.
B.T.U. per B.H.P. hour	10030	10910	15050	
Brake kinetic efficiency	25.37	23.32	16.9	B.H.P. basis
Mechanical efficiency	89	87.5	82.5	Estimated
Indicated kinetic efficiency	28.5	26.7	20.5	I.H.P. basis
Speed variation, maximum	4.2%	.....	....	No load speed 206.6
Speed variation, no to full load	....	1.8%	....	
Speed variation, no to half load.	....	......	0.7%	
Per cent full rating	138.4%	101.6%	51.6%	On producer gas

* Pittsburg natural gas—Junker calorimeter. Engines rated 235 B.H.P. on 130 B.T.U. producer gas.

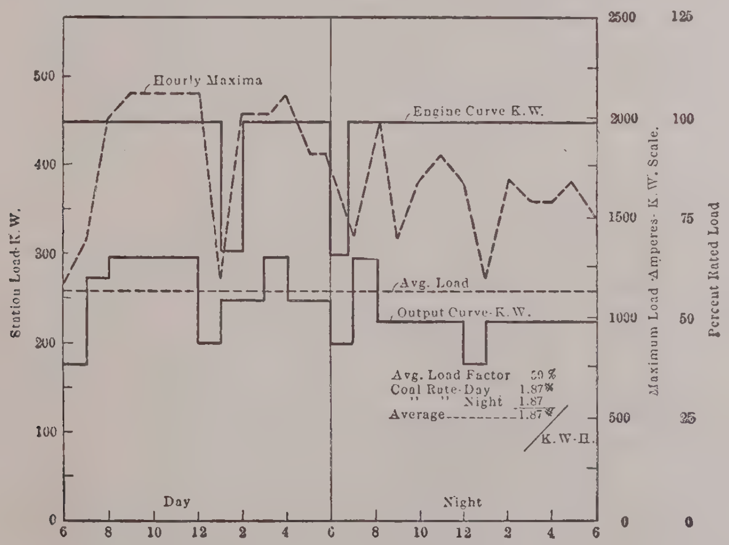


FIG. 574.—Load Curves, Plant of the Gould Coupler Co., Depew, N. Y.

After installation, careful daily records were kept in the plant, from which a good idea of plant efficiency may be obtained. Table 75 shows one of the logs typical of those kept from day to day.

TABLE 75

TYPICAL LOG. GOULD COUPLER COMPANY, DEPEW, N. Y.

GAS POWER HOUSE DAILY STATEMENT FOR SEPTEMBER 26, 1905

Time.	Producers in			Engines in			Pumps in			Station Watt-meters, Output kw.	Volts.	Maximum Amperes.	Remarks.
	1	2	3	1	2	3	1	2	3				
A.M. 7	△	△		△	△	△	△	△	△	1459800	230	1000	
8	│	│		│	│	│	│	│	│	300	230	2100	8% overload
9	│	│		│	│	│	│	│	│	300	230	2100	
10	│	│		│	│	│	│	│	│	200	230	2000	
11	│	│		│	│	│	│	│	│	300	230	2100	
12	│	│		▽	▽	│	│	│	│	300	230	2100	
P.M. 1	│	│		△	△	│	│	│	│	100	230	2000	
2	│	│		│	│	│	│	│	│	300	230	2100	
3	│	│		│	│	│	│	│	│	200	230	2000	
4	│	│		│	│	│	│	│	│	300	230	2000	
5	│	│		│	│	│	│	│	│	300	230	2000	Full load about 1950 amperes
6	│	│		│	│	│	│	│	│	200	230	1800	
7	│	│		│	│	│	│	│	│	250	230	1700	
8	│	│		│	│	│	│	│	│	300	230	1900	
9	│	│		│	│	│	│	│	│	300	230	1900	
10	│	│		│	│	│	│	│	│	300	230	1800	
11	│	│		│	│	│	│	│	│	300	230	1900	
12	│	│		│	▽	│	│	│	│	300	230	1900	
A.M. 1	│	│		│	△	│	│	│	│	150	230	1100	
2	│	│		│	│	│	│	│	│	300	230	1900	
3	│	│		│	│	│	│	│	│	350	230	1800	
4	│	│		│	│	│	│	│	│	300	230	1800	
5	│	│		│	│	│	│	│	│	300	230	1900	
6	▽	▽		▽	▽	▽	▽	▽	▽	250	230	1700	

METER READINGS:

Day Engineer	Hanley
" Helper	Young
" Producer Man	Benarak
" " "	Lea
Night Engineer	Lowe
" Helper	Runer
" Producer Man	Smith
" " "	——

6.00 p.m.	1462600
Previous	1459700
Output	2900

Coal, 6000; Coke, ——
Rate 2.06 lbs. per kw. hour

6.00 a.m.	1466000
Previous	1462600
Output	3400

Coal, 6300; Coke, ——
Rate, 1.85 lbs. per kw. hour

Since the above observations were made the plant has been giving much better efficiency, the coal consumption averaging 1.8 lbs. per K.W. hour with an 85% loading factor. This corresponds to a plant efficiency of over 15.4% at the engine shaft, or 14% at the switchboard. Several runs averaging 1.55 lbs. per K.W. are recorded, equivalent to a plant efficiency of 17.7% at the shaft, or 16.3 % at the switchboard.

The log needs no further explanation. From the daily records thus kept Table 76, covering a period of ten consecutive days, has been compiled. The term "loading factor" used represents the ratio of average twenty-four hour station load to rated capacity. The nature of the varying load on any one of these days is well shown in Fig. 574 for Sept. 25th.

Table 76

OPERATING DATA

450 K.W. Producer Gas Power Plant

Date, September, 1905.	Engine Hours Run.	Per Cent Full Output.		Load K.W.		Station Loading Factor.*	Fuel Used.	Coal per K.W.	Coal per B.H.P. Hour.
		Day Run,	K.W.	Average.	Rated.				
20	71	98⅔	4850	202	450	45	11100	2.29	....
21	65.5	91	5275	220	450	49	11400	2.16	....
22	70	97¼	6550	273	450	61	12700	1.93	....
23	45.75	63½	4025	188	450	41.7	8800	2.18	....
24 Sun.	24	33⅓	2250	187.5	450	41.6	6600	2.93†	....
25	70.5	98	6400	267	450	59	12000	1.87	....
26	69.75	97	6300	263	450	58	12300	1.95	....
27	70.5	98	6700	279	450	62	12600	1.88	....
28	72.0	100	6700	279	450	62	12900	1.92	....
29	70.5	98	6600	275	450	61	12900	1.95	....
Average . .	63	87¼	5565	243.4	450	54	11330	2.04	1.44

Westinghouse vertical three-cylinder engines—Loomis-Pettibone producers.

* Loading factor = per cent continuous generating capacity.

† Includes extra coke used on Sunday for starting new fire.

This plant offered an unusually good opportunity of observing the effect of load factor upon plant efficiency. The results of this investigation are given by Mr. Bibbins in Fig. 575. These curves were constructed from data obtained since the main tests were made and show a considerable improvement over the figures then obtained. This was due to more careful operation, but largely also to increased loading of the plant.

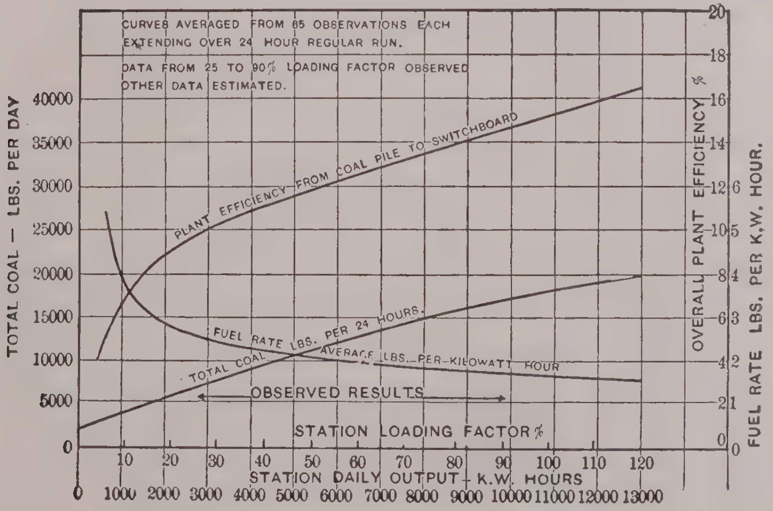


Fig. 575.—Effect of Load Factor on Plant Efficiency.

TABLE 77

HOLDER DROP TESTS

SUMMARY OF RESULTS

Test No.	*A*	*B*	*C*	*D*	*E*	Remarks.
Duration of Run, Minutes	11	8	10	10	10	
Load, per cent engine rating	no load	25.4	45.1	70.6	102.2	Circum. holder 110.33 ft.
Brake horse-power	......	127.0	225.5	353.0	511.5	
Kilowatts	......	84.1	154.3	243.5	352.0	
Speed, revolutions per min.	158.00	156.0	154.0	152.0	150.0	Barom. = 29.26″
Holder drop, feet per hour	16.91	24.96	32.22	39.89	51.60	Av. temp. of gas, 71.6° F.
Cubic feet per hour, 30′ 60°	15760.00	23270.00	30050.00	37280.00	48200.00	Av. gas pressure, 2⅞ in. water
Gas consumption rate:	(Std. gas.)					
Cubic feet per B.H.P. hour	......	183.2	133.2	105.5	94.25	Correction factor—0.9642
Cubic feet per K.W. hour	......	276.8	194.8	153.1	137.0	
Heat value of gas: *a*						
Effective B.T.U. per cu.ft.	106.4	106.4	106.4	106.4	106	*a* Av. of all tests
B.T.U. per B.H.P. hour	......	19480.00	14160.00	11215.00	10030.00	
B.T.U. per K.W. hour	......	29430.00	20720.00	16280.00	14560.00	
Thermal efficiency brake. . %	......	13.05	17.96	22.68	25.36	
Thermal efficiency (electrical) %	......	11.6	16.46	20.96	23.42	

TABLE 78

FRACTIONAL LOAD EFFICIENCIES

HOLDER DROP TESTS

Nominal Load.	¼	½	¾	Full.	Overload.
Load, brake horse-power	125.00	250.00	375.00	500.00	550.00
Gas consumed,* cubic feet per B.H.P. hour	190.00	127.00	105.6	95.00	92.20
Heat consumed, B.T.U. per B.H.P. hour	20210.00	13510.00	11240.00	10100.00	9800.00
Heat consumed,* B.T.U. per K.W. hour	30530.00	19700.00	16340.00	14675.00	14300.00
Heat consumed,* B.T.U. per I.H.P. hour	11180.00	10600.00	9050.00	8460.00	8295.00
Thermal efficiency, per cent brake	12.58	18.84	21.66	25.21	25.97
Thermal efficiency, per cent elec.	11.16	17.32	20.9	23.25	23.85
Thermal efficiency, per cent indicated	22.75	24.1	28.14	30.1	30.7

EQUIVALENT COAL CONSUMPTION † FOR VARIOUS PRODUCER EFFICIENCIES, POUNDS PER UNIT HOUR

Producer efficiency:						
100%,	brake horse-power hour	1.413	0.944	0.785	0.705	0.685
	kilowatt hour	2.13	1.376	1.141	1.025	0.999
80%,	brake horse-power hour	1.766	1.181	0.980	0.882	0.857
	kilowatt hour	2.663	1.720	1.426	1.281	1.250
70%,	brake horse-power hour	2.015	1.347	1.120	1.006	0.977
	kilowatt hour	3.040	1.964	1.63	1.465	1.426

---

* Assuming same coal used on test—14 321 B.T.U.

† Standard gas—106.4 B.T.U. (effective), 60° 30″ Hg.

TABLE 79

FIFTY-ONE HOUR TEST—SIX-HOUR AVERAGES. CORRECTED DATA ONLY

PERIOD.	JUNE 24, P.M.		A.M. JUNE 25. P.M.				A.M. JUNE 26. P.M.			TOTAL AVERAGE.
	3–6	6–12	12–6	6–12	12–6	6–12	12–6	6.12	12–6	51 Hours.
LOAD										
K.W. by wattmeter	303.6	358.9	347.25	333.1	320.3	354.5	338.85	322.0	319.0	332.2
Kilo-volt-amperes	306.2	354.9	352.9	336.4	306.2	335.3	338.2	317.7	310.7	328.72
B.H.P. from K.W.	440.7	522.2	504.8	483.8	451.5	487.4	492.2	467.3	462.8	482.3
B.H.P. from Kilo-volt-amperes	445.0	516.4	512.7	498.5	445.0	487.0	491.5	461.0	451.0	477.31
Amperes	1290.0	1412.0	1411.0	1382.0	1250.0	1341.0	1353.0	1302.0	1276.0	1335.2
Volts	237.3	251.1	250.1	243.5	245.0	250.0	250.0	244.0	243.3	246.03
COAL.										
Pounds fired per hour	443.5	454.7	455.3	466.0	475.0	463.8	449.7	522.0	452.0	466.0
Pounds fired per square foot per hr.*	12.71	13.04	13.06	13.36	13.63	13.30	12.90	14.97	12.96	13.36
WATER.										
Cubic feet per hour	613.3	594.1	613.3	595.3	592.6	653.1	652.0	651.0	677.0	626.8
Cubic feet K.W. hour	2.126	1.625	1.765	1.786	1.850	1.843	1.925	2.020	2.120	1.886
Gallons per B.H.P. hour	10.43	8.53	9.100	9.200	9.260	10.04	9.270	10.44	10.95	9.740
Average inlet temperature °F.	65.00	65.50	65.00	65.40	66.30	67.00	66.00	66.60	67.30	66.01
Average temperature rise °F.	42.80	54.60	47.70	48.90	49.80	46.00	46.60	43.80	43.60	47.10
GAS.										
Heat value by calorimeter	112.3	109.16	108.0	116.7	119.4	113.5	111.8	119.3	120.9	114.56
Heat value by analysis	....	119.74	105.9	110.4	109.78	108.5	113.5	116.13	115.13	112.40
Heat value, maximum	123.2	122.60	122.1	125.8	125.6	117.1	116.3	122.8	126.0	126.00
Heat value, minimum	105.0	108.10	101.5	101.9	111.6	109.8	107.2	110.2	111.4	101.50

* Rate of gasification per square foot of fuel bed area of producers.

Calorific values all reduced to effective at 62° F. 30-inch Hg.

Horizontal Engine. (*a*) Tests made by Messers Alden and Bibbins on a 23½×33″, 500-B.H.P. horizontal engine direct-connected to a 300-K.W. generator in the plant of the Norton Company at Worcester, Mass., during June, 1907, and reported to the American Society of Mechanical Engineers, December, 1907.

The fuel used was producer gas made in Loomis-Pettibone pressure producers. The coal fired in regular operation was Clearfield bituminous, nine samples of which averaged: Volatile matter, 19.87%; fixed C., 73.71; moisture, .87; ash, 5.54; B.T.U. per pound dry, 14 450 B.T.U.; as received, 14 321. The composition of the hard coal used to start fires was: Volatile matter, 5.20%; fixed C., 78.95; moisture 3.20; ash, 12.65; B.T.U. per pound dry coal, 12 709; B.T.U. as received, 12 320. The average sulphur in the Clearfield coal was .83%.

The tests made were of two kinds, first a 51-hour continuous test to determine the coal, water and oil consumption, and the mechanical efficiency; and second, a series of holder drop tests to determine the net heat consumption at various loads. It should be noted that the 51-hour test was a service run under the regular operating conditions with the plant in charge of the regular force.

The main result of the holder drop tests are presented in the upper part of Table 77. These results were plotted, and from these curves were obtained the figures in the lower part of the table. The same curves, by interpolation, also gave the figures in Table 78. The reduction from electrical output to brake horse-power was made by aid of the generator efficiency curve.

Table 80

FIFTY-ONE HOUR TEST

Summary of Results (Norton Co., June 24–26, 1907)

	Load, Kilowatt.	Water, Cubic Feet.	Oil, Gallons.	Coal,* Pounds.
Quantity at finish	363550.0	94900.0	2.875	23775
Quantity at start	345710.0	63560.0	.....	.....
Difference	16840.0	31340.0	2.875	23775
Correction	+117.3	....	....	.....
Corrected difference	16957.3	31340.0	2.875	23775
Elapsed time	51 hrs.	50 hrs.	48 hrs.	51 hrs.
Rate per hour	332.5	626.8	0.06	466

	Water, Cubic Feet.	Water, Gallons.	Oil, Gallons.	Coal, Pounds.
Rate per K.W. hour (332.5 K.W.)	1.885	14.12	0.00018	1.402
Rate per B.H.P. hour (482.9 B.H.P.)	1.3	9.74	0.000125	0.965
Rate per I.H.P. hour (579.0 I.H.P.)	1.078	8.075	0.000104	0.805

* Clearfield run-of-mine—14 321 B.T.U. per pound as fired. Average thermal efficiency of plant 18.43%; engine, 24.93%; producer, 73.81%. Average gasification rate, 13.36 lbs. per sq.ft. per hour.

For the 51-hour test Table 79 gives the consecutive 6-hour averages, while Table 80 shows the total average readings. The load during this test was fairly uniform and as near rating as the demand of the mill would permit. During the night shifts it was often possible to exceed rating, thus on the night of June 24 the average load was 522 B.H.P. for six hours, which corresponds to an overload of nearly 20% on the generator. At the same time the heating value of the gas was below the average, only 109 B.T.U. per cu.ft. Fig. 576 shows a typical set of cards taken during the long run. This is one of the sets taken once an hour, together with the electrical readings, to determine the mechanical efficiency of the engine. From an investigation of these cards it appears that the mechanical efficiency at the average load carried was 83.5%, and at full load, 83.8%. These figures include all pump work.

Additional determinations on oil and water consumption were summarized by the observers as follows:

Average water consumption, for engine only, 9.74 gallons per B.H.P. hour with 66° F. inlet temperature and 47.1° F. rise.

Average cylinder oil consumption, 1.44 gallons per 24 hours, equivalent to 0.6 gallon per operating day, or 3.2 gallons per operating week.

Average producer efficiency, 74.4% at full load, 73.8% at average test load—both based upon lower or effective heat value of gas.

Producer gas, average, 114.6 effective B.T.U. during 51-hour test; maximum variation 11.5% above and below mean. Difference between total effective heat values—about $4\frac{2}{3}$%.

This important test, complete in all respects, was supplemented by speed variation tests, the results of which are given in Table 81.

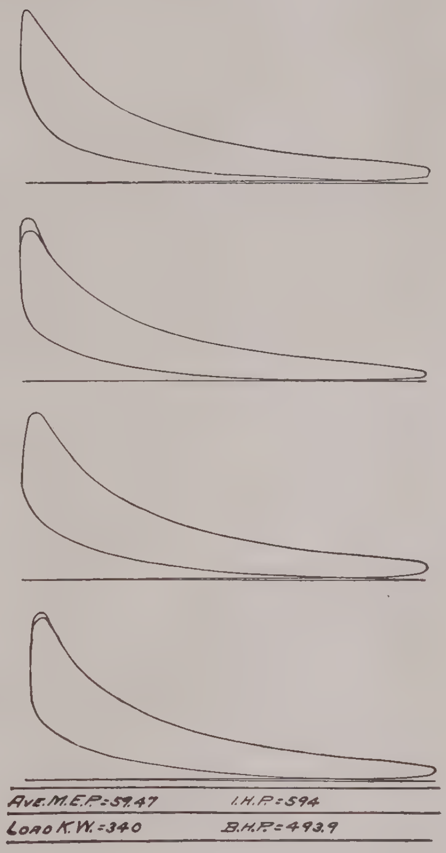


Fig. 576.

TABLE 81

SPEED VARIATION TESTS

Speed, revolutions per minute	155	154.0	152.0	150.0	149.0	148.0
Volts	255	255.0	257.0	258.0	258.0	257.0
Amperes	....	327.5	665.0	955.0	1303.0	1347.0
K.W.	....	86.1	170.8	246.6	336.1	346.0
B.H.P.	....	129.6	247.6	356.5	489.3	503.0
Per cent full rating	....	25.9	49.5	71.2	97.9	100.5
Speed drop, per cent ± mean	....	0.819	0.958	1.597	1.916	2.236

INSTANTANEOUS LOAD TEST JUNE 27, 1907, 6 P.M.

No load to full load, 280 volts, 1190 amperes, 345 kilowatts, 502 brake horse-power

No-load speed	155 revolutions per minute
Load thrown on	148 revolutions per minute
Load thrown off	155 revolutions per minute
Difference	7 revolutions per minute
Speed variation	4.6% of total – 2.3% ± mean speed

Other data available on the operation of this plant is given in Table 82, which was published in *Power* for July, 1907. This table gives the average operating results for seven consecutive weeks beginning with Jan. 6, 1907. The average coal consumption per B.H.P. or per K.W. hour for this entire period is some 25% higher than in the 51-hour test made by Messrs. Alden and Bibbins in June of the same year. This is largely due to the fact that the loading factor during this time was only about three-fourths rating, while during the June tests it was practically unity. A slightly lower grade of coal may also have affected the results to a small extent.

TABLE 82

SUMMARY OF OPERATING DATA OF GAS-POWER PLANT OF THE NORTON COMPANY, WORCESTER, MASS.

Week Ending.	Hours Run.	Output, K.W. Hours.	Coal Burned, lbs.*			Coal per K.W. Hour.	
			Producer.	New Fires.	Total.	Producer.	Total.
January 6	45–5	10150	20690	1400	22090	2.03	2.17
January 13	50–5	11900	24600	2700	27300	2.07	2.29
January 20	55	13200	24530	2570	27100	1.86	2.05
January 27	54–40	12400	22500	2000	24500	1.84	2.00
February 3	55	13600	23800	2400	26200	1.75	1.93
February 10	55	13000	24265	3060	27325	1.87	2.10
February 17	55	13000	23700	‡2500	26200	1.82	2.01
Total	369–50	87250	164085	16630	180715	. . . .	. . . .
Average	53	12464	23441	2376	25816	1.88	2.07

Week Ending.	Generator Efficiency.	Coal per B.H.P. Hour.		Heat Consumption.†		Plant Efficiency.†		Station Loading Factor.	Average Load per Cent Engine Rating.
		Producer.	Total.	B.T.U. per K.W. Hour.	B.T.U. per B.H.P. Hour.	Electrical.	Brake.		
January 6	92	1.39	1.49	28400	19500	12.00	13.1	20.2	75.0
January 13	92	1.42	1.57	29000	19900	11.75	12.9	23.6	79.0
January 20	92	1.28	1.41	26000	17900	13.10	14.2	26.2	80.0
January 27	92	1.26	1.37	25800	17600	13.20	14.5	24.3	74.8
February 3 . . .	92	1.20	1.32	24500	16800	13.90	15.2	27.0	82.4
February 10 . . .	92	1.28	1.44	26200	17900	13.00	14.2	25.8	78.8
February 17 . . .	92	1.25	1.38	25500	17500	13.40	14.5	25.8	78.8
Total	. .	. . . .	. . . .	. . . . .	. . . . .	. . . . .	. . . .	. . . .	. . . .
Average	92	1.29	1.42	26300	18050	12.96	14.1	21.9	78.7

* Med. grade Penn. bituminous coal up to January 21. Pocahontas slack thereafter. Assumed, 14000 B.T.U. average.

† For continuous running, not including 10% loss in pulling fire weekly.

‡ Estimated from previous runs.

(*b*) Tests made under the direction of Mr. W. Dalton, Chief Engineer, on a horizontal tandem gas engine in the plant of the American Locomotive Co. at Richmond, Va., and reported by J. R. Bibbins to the Convention of the American Institution of Electrical Engineers, June, 1908.

The engine is of the usual double-acting tandem construction, with cylinder diameter equal to 23½″ and stroke equal to 33″. The fuel used in the producers was bituminous coal. The producer is an R. D. Wood induced draft bituminous type with water-sealed bottom and centrifugal type tar extractor. An internal belt evaporator located above the fire bed furnishes the vapor required for dissociation. Wet and dry scrubbers are also used in addition to the centrifugal tar extractor. A large gasometer serves as a mixing chamber and provides ample storage for a short period of producer shut down, as well as for starting up the entire plant.

The demand on the plant is fairly constant owing to the overlapping power requirements of a considerable number of electrically driven machines throughout the works. The test was continued for practically four weeks, the first half of this time, 223 hours, at nearly full load, and the remaining two weeks on ¾ load and ½ load. The observations were made entirely by the staff of the locomotive company in order to determine the fulfilment of guarantees.

The principal results of the test are shown in the following table:

TABLE 83

GENERAL RESULTS OF TEST

	Full Load.	¾ Load.	½ Load.
Length of run, hours	223	125	136
Average load, kilowatts	312.3	228.3	159.6
Average load, computed B.H.P.	455.0	333.0	238.0
Load, per cent engine rating	91.0	67.6	47.5
Load, per cent generator rating	104.0	77.2	53.2
Coal gasified, lbs.	115289	54143	47775
Coal gasified per hour	517.0	433.0	351.0
Output, kilowatt hour	69650	28540	21710
Pounds coal, per kilowatt hour	1.654	1.697	2.20
Pounds coal, per kilowatt hour guaranteed	1.93	2.10	2.64
Pounds coal, per B.H.P. hour	1.14	1.31	1.56
Average heat value of coal, B.T.U.	14392	14392	14392
B.T.U. per kilowatt hour	23700	27280	31650
B.T.U. per B.H.P. hour	16415	18710	21670
Per cent thermal efficiency, brake	15.51	13.6	11.75
Per cent thermal efficiency, electric	14.35	12.65	10.78

Coal.—Pocahontas run-of-mine; average heat value dry sample, 14 703, as fired, 14 392; volatile matter, 22.8%, ash, 4.5%, sulphur, 1%.

Test.—Aug. 12, 7 A.M. to Sept. 7, 12 M.

At a load of 312 K.W. the engine is running slightly under rated load, and, on the basis of the figures observed, it is estimated that the coal consumption at full load, 350 K.W., would have been 1.59 lbs. per K.W., or 1.09 lbs. per B.H.P.

During the four weeks test there were of course several periods of idleness, as the plant is not operated over Sunday. The producer operator estimates that from 6 P.M. on Saturday to 7 A.M. on Monday 1700 lbs. of coal on the average were required to cover stand-by losses.

Determinations of oil and water consumption were also made during the 223-hour test. The average consumption of cylinder oil was .09 gallons per hour, equivalent to 4.9 gallons per week for a 10-hour working day. The average consumption of cooling water was 6.56 gallons per B.H.P. hour at heavy load, the inlet temperature varying from 75–80° F., and the outlet from 140–150°.

2. De La Vergne Machine Co., New York. This firm builds four distinct types of engines, the Körting 2- and 4-cycle engines, which they now call type KT and type KF, the Hornsby-Akroyd 4-cycle horizontal oil engine (type HA) and the De La Vergne 2-cycle vertical oil engine (type S). (Two other oil engines, to be known as types FH and FV, horizontal and vertical respectively, have also been produced in which heavy crude and fuel oils are completely burnt with the assistance of highly compressed air, but these two types are at this time (Dec., 1908) only emerging from the experimental stage. They are omitted in this edition.)

The first two of the engines above mentioned have already been described in some detail on p. 334 to 341, and hence it will only be necessary to point out the differences in the American types where they exist.

TABLE 84

ENGINE.	SINGLE.	TWIN.	SINGLE.	TWIN.
B.H.P. with producer and blast-furnace gas	500	1000	1000	2000
B.H.P. with natural gas	600	1200	1200	2400
Power cylinder	$25\frac{1}{2}''\times45''$		$38''\times60''$	
Revolutions per minute	100		70	
Piston speed, feet per minute	750		700	
Main bearings	$12\frac{5}{8}''\times23\frac{5}{8}''$		$18''\times33''$	
Outboard bearing	$12''\times21''$	....	$14\times25\frac{1}{4}''$	....
Main crank pin	$12\frac{5}{8}''\times13\frac{3}{4}''$		$18''\times19\frac{1}{4}''$	
	two	four	two	four
Main cross-head pins	$8\frac{1}{4}''\times7\frac{7}{8}''$		$12\frac{3}{4}''\times11\frac{1}{2}''$	
Main cross-head shoe bearing surface	$19\frac{3}{4}''\times34\frac{5}{8}''$		$27\frac{1}{2}''\times52''$	
Main piston rod, diameter	$6\frac{1}{2}$		10	
"Light" fly-wheel, factory work, diameter	17′.6		22′	
" " " , " " , weight	36000		66000	
"Heavy" " , electric " , diameter	17′ 10″		23′	
" " , " " , weight	72000		130000	
Bore of fly-wheels	16″		$23\frac{5}{8}''$	
Diameter pumps for furnace gas	$28\frac{3}{8}''$		$38\frac{1}{4}''$	
Diameter air pump for producer and natural gas	$28\frac{3}{8}''$		$38\frac{1}{4}''$	
Diameter gas pump for producer and natural gas	$25\frac{3}{4}''$		$34\frac{5}{8}''$	
Stroke of all pumps	$33\frac{1}{2}''$		$49\frac{1}{4}''$	
Diameter gas inlet pipe	13″		18″	
Diameter exhaust pipe	17″		24″	
Diameter cooling water pipe	$3\frac{1}{2}''$		5″	
Length over all (heavy fly-wheel)	39′ 4″		55′ 11″	
Width over all *	23′ 5″	33′ 6″	30′	45′
Shaft center to floor level	34″+6″		39″+7″	
Shipping weight without wheel	132000	250000	380000	746000

* Including extended shaft, or generator between the cylinders of a twin.

Table 85

PRINCIPAL DIMENSIONS OF 4-CYCLE SINGLE-ACTING GAS ENGINES (Körting Patents)

Single-cylinder Engines

Rated B.H.P. with producer gas, 82% and 68 lbs. M.E.P.	75	85	100	125	150	175
Do. with natural or illuminating gas,* 78 lbs. M.E.P.	85	100	115	145	175	200
Revolutions per minute, Std.	170	160	155	145	135	130
Diam. and stroke of motor cylinder, in millimeters	430×700	460×755	490×800	540×876	590×955	650×955
Do., in inches	$16\frac{15}{16}\times 27\frac{9}{16}$	$18\frac{1}{8}\times 29\frac{3}{4}$	$19\frac{1}{4}\times 31\frac{1}{2}$	$21\frac{1}{4}\times 34\frac{1}{2}$	$23\frac{1}{4}\times 37\frac{5}{8}$	$25\frac{5}{8}\times 37\frac{5}{8}$
Piston speed, feet per minute	780.8	792.6	813.6	833.5	846.0	814.6
Crank pin, diameter×length	$7\frac{7}{8}\times 8\frac{11}{16}$	$8\frac{7}{16}\times 9\frac{7}{16}$	$9\frac{1}{16}\times 10\frac{1}{4}$	$10\frac{1}{16}\times 11$	$11\times 12\frac{3}{16}$	$12\frac{3}{8}\times 14\frac{9}{16}$
Wrist or piston pin (bearing, diameter×length	$5\frac{9}{16}\times 8\frac{1}{8}$	$5\frac{15}{16}\times 8\frac{11}{16}$	$6\frac{1}{8}\times 9\frac{7}{16}$	$6\frac{11}{16}\times 10\frac{1}{4}$	$7\frac{1}{2}\times 11$	$9\frac{7}{16}\times 12\frac{5}{8}$
Main bearings (2), diam.×length	$7\frac{11}{16}\times 14\frac{3}{4}$	$8\frac{1}{4}\times 16\frac{3}{16}$	$8\frac{7}{8}\times 17\frac{15}{16}$	$9\frac{7}{8}\times 18\frac{5}{16}$	$10\frac{5}{8}\times 19\frac{7}{8}$	$11\frac{13}{16}\times 21\frac{5}{8}$
Outboard bearing (1), diam.×length	$6\frac{1}{8}\times 12\frac{3}{16}$	$6\frac{1}{8}\times 12\frac{3}{16}$	$6\frac{1}{8}\times 12\frac{3}{16}$	$9\times 27\frac{1}{4}$	$9\frac{1}{2}\times 23$	$10\frac{1}{2}\times 31\frac{1}{2}$
Diameter of heavy fly-wheel	10′ 8″	11′ 3″	12′	12′ 7½″	13′ 5½″	14′
Diameter of light fly-wheel	10′ 11″	10′ 10″	12′ 4½″	12′ 6″	13′ 5½″	13′ 9″
Bore of (either) wheel or of pulley	$9\frac{7}{16}$	$10\frac{1}{16}$	$10\frac{5}{8}$	$11\frac{5}{8}$	$12\frac{5}{8}$	$14\frac{3}{16}$
Weight of heavy fly-wheel,† net lbs.	16700	17200	21000	27200	40600	50200
Weight of light fly-wheel, net lbs	9900	11400	13200	17600	24000	30000
Weight of engine without wheel, gross	24000	28800	34600	42800	48200	50300
Weight of engine without wheel, net	22500	26500	31500	39100	43900	45600
Driving pulley (split), diameter	4′ 6″	5′	6′	7′	9′	10′
Driving pulley width of double belt, in.	12	14	14	16	16	18
Length over all incl. heavy wheel	17′	18′ 8″	19′	21′	22′ 4″	23′ 7″
Width over all incl. outboard bearing.	9′ 2″	11′ 9″	12′ 1″	11′ 9″	12′ 4″	13′ 10″
Height, bottom of frame to top of heavy wheel	7′ 8″	8′ 3″	8′ 8″	9′ 3″	9′ 9″	10′ 2″
Height, bottom of frame to center of crank-shaft	25⅝″	26¾″	28⅜″	28⅜″	28″⅜	31½″
Weight of frame casting, rough, lbs.	8750	10000	10780	12175	16075	23340
Size of air pipe, inches	7	7	7	8	8	10
Size of exhaust pipe, inches	7	7	8	8	9	12
Size of producer gas pipe, inches	7	7	7	7	7	8
Size of natural or illum. gas pipe, in.	4	4	4	5	5	6

* With blast-furnace gas, M.E.P. = 60 lbs. per sq.in. and B.H.P. is but 88.2% that with producer gas.

† *Light fly-wheels* are for general power engines, pumping, etc. *Heavy wheels* are for D.C. direct-connected or belt-driven generators of any kind. Direct-connected A.C. generators require twin engines and *extra heavy fly-wheels.* All wheels made in halves.

Table 86

TWIN-CYLINDER ENGINES

Rated B.H.P. with producer gas, 68 lbs.	150	170	200	250	300	350
Do. with natural or illuminating gas, 78 lbs. M.E.P.	170	200	230	290	350	400
Rev. per min. (same as single cyl.)	170	160	155	145	135	130
Size of cyls. (same as single cyl.), mm.	430×700	460×755	490×800	540×876	590×955	650×955
Width of engine,* including a generator placed at end of crank-shaft	17′	19′	21′	22′ 6″	23′ 6″	25′
Do. excluding the above generator	13′	14′	15′	16′	17′ 3″	18′ 6″
Do. with generator or pulley in middle	15′ 9″	17′ 8″	18′ 2″	18′ 4″	19′ 8″	22′ 1″
Weight of whole engine without fly-wheel or pulley, gross lbs	46500	55800	66600	83000	93000	97500
Do. net lbs.	43600	51400	61000	75800	85200	88500
Driving pulley (split), diameter	6′	7′	7′ 6″	9′	10′	12′
Driving pulley, width of double belt, in.	18″	20″	22″	24″	28″	28″

* In each case is the fly-wheel assumed as between the two cylinders.

Note.—Fly-wheel and other parts are alike for twin- and single-cylinder engines.

1. *The Körting 2-cycle engine.* (Type KT.) The main features of the American design of this engine are shown in Fig. 577. This picture, however, does not show the latest governing gear which is illustrated in detail in Figs. 578 and 579. The former indicates that the air pump and its valves operate in the ordinary way. The gas pump cylinder, however, is provided with a ring of ports at the middle so that for one-half of the discharge stroke the piston forces the gas back into the suction main. Each end of the gas pump is fitted at the bottom with suction and discharge valve and at the top with a by-pass valve. This valve is positively operated by eccentric and reach rods over the cylinder, as shown in Fig. 578. It is tripped by a gear similar to a Corliss gear, Fig. 579, at a point in the second half of the gas pump discharge stroke determined by the governor. The method of operation of this gear becomes clear from a study of the figure. As long as this valve remains open, the gas continues to be forced back into the suction main. As soon as the gear trips, however, the spring shown closes the valve and the gas is forced through the discharge valve at the bottom of the pump and into the power cylinder for the remainder of the pump stroke.

The table on p. 408 shows the principal dimensions of De La Vergne-Körting 2-cycle gas engines.

2. *The Körting 4-cycle Engine.* (Type KF.) This engine has already been described (p. 334) and there is nothing radically different in the American design. The tables on page 409 give the principal dimensions of the various sizes made by the De La Vergne Co., together with some other data.

The features of the De La Vergne suction producer used with these 4-cycle engines are in general those of the Körting suction producer already described on p. 274.

The following table gives the principal dimensions required for electric power plants equipped with De La Vergne type KF engines and producers. The letters refer to Figs. 580 and 581, which show two lay-outs, one for direct connected, the other for belted units.

TABLE 87

ELECTRIC POWER PLANTS WITH DE LA VERGNE GAS ENGINES AND SUCTION GAS PRODUCERS

Arranged for Single-cylinder Units of 75 to 175 B.H.P. Capacity

A. Gas engines (Körting type).
B. Gas producers.
C. Vaporizers.
D. Coke scrubbers.
E. Sawdust purifiers.
F. Expansion tanks.
G. Electric motor, for
H. Belt-driven air compressor.
I. Compressed air receiver.
J. Electric motor, for
K. Blower for producers.
L. Exhaust silencers.

TABLE OF APPROXIMATE DIMENSIONS IN FEET

Size Units, B.H.P.	Direct Connected (Fig. 580).						Belted (Fig. 581).					
	75	85	100	125	150	175	75	85	100	125	150	175
M	22	23	24	25.5	27	28.5	15	16	17	17	18	19
N	31	33	34	35	37	40	38	38	40	41	42	43
O	17	17	18	18	19	19	15	15	15	15	16	16
P	20	21	22	23.5	25	26.5	22	23	24	25.5	27	28
Q	24	24	28	29	32	32	13	13	15	15	18	18
R	15	15	15	15.5	16	16	..	..	..	..	..	..

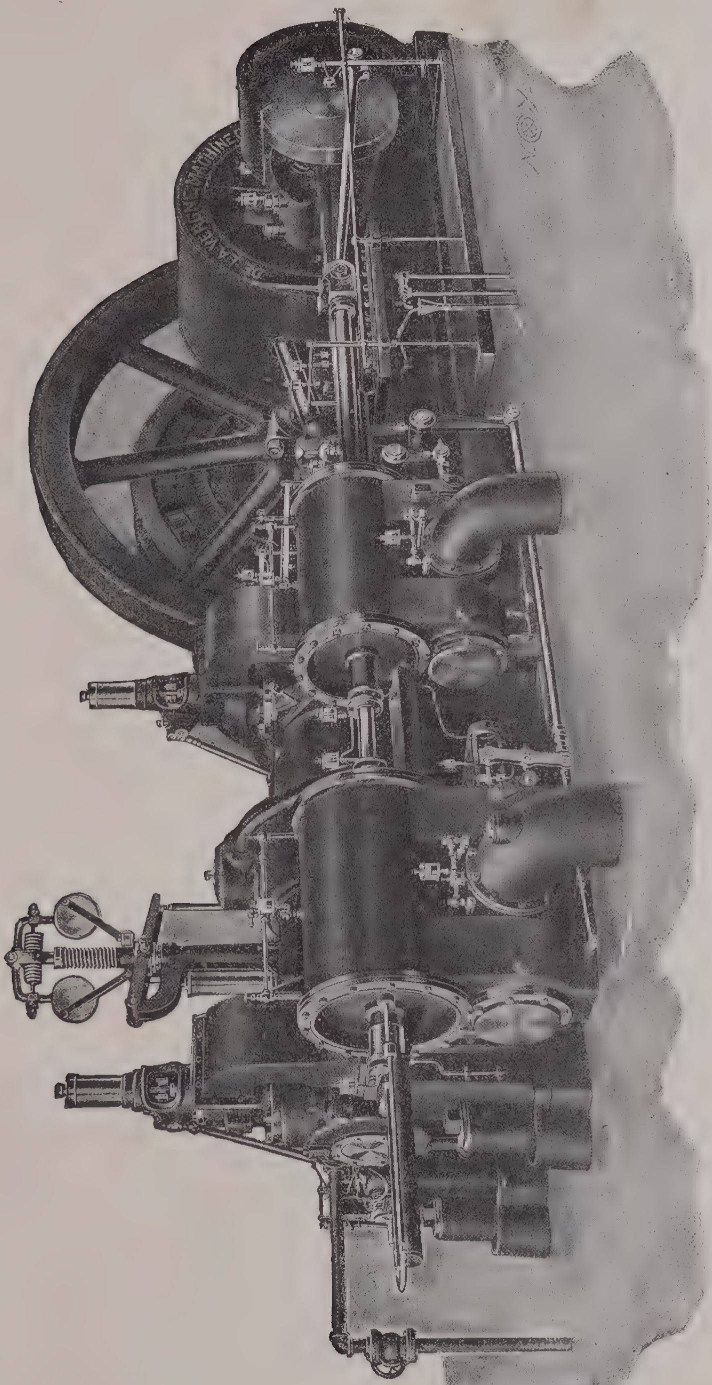


FIG 577.

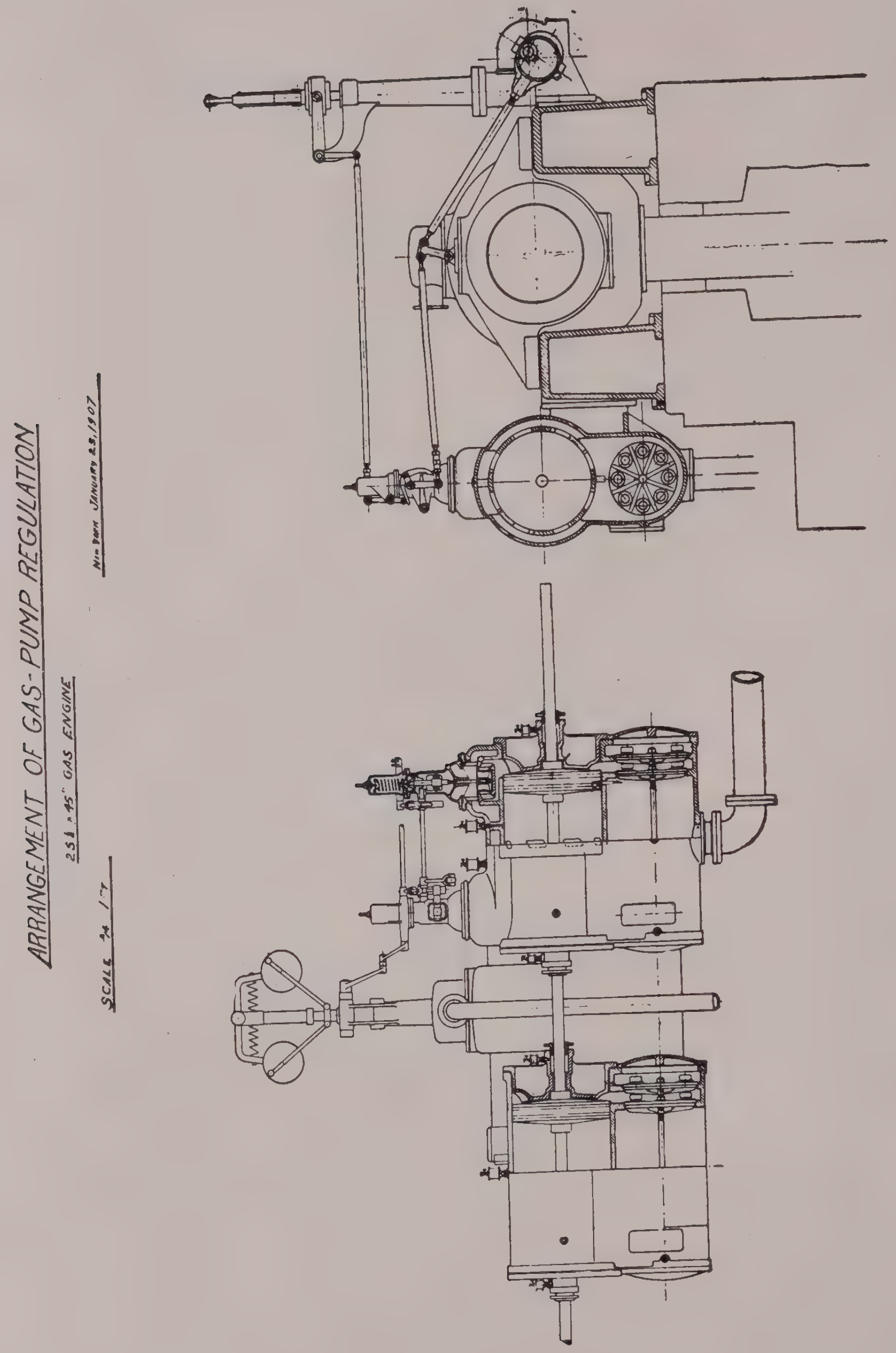

ARRANGEMENT OF GAS-PUMP REGULATION
25½ x 45" GAS ENGINE
SCALE ¾ 1'
NEW YORK JANUARY 25, 1907

Fig. 578.

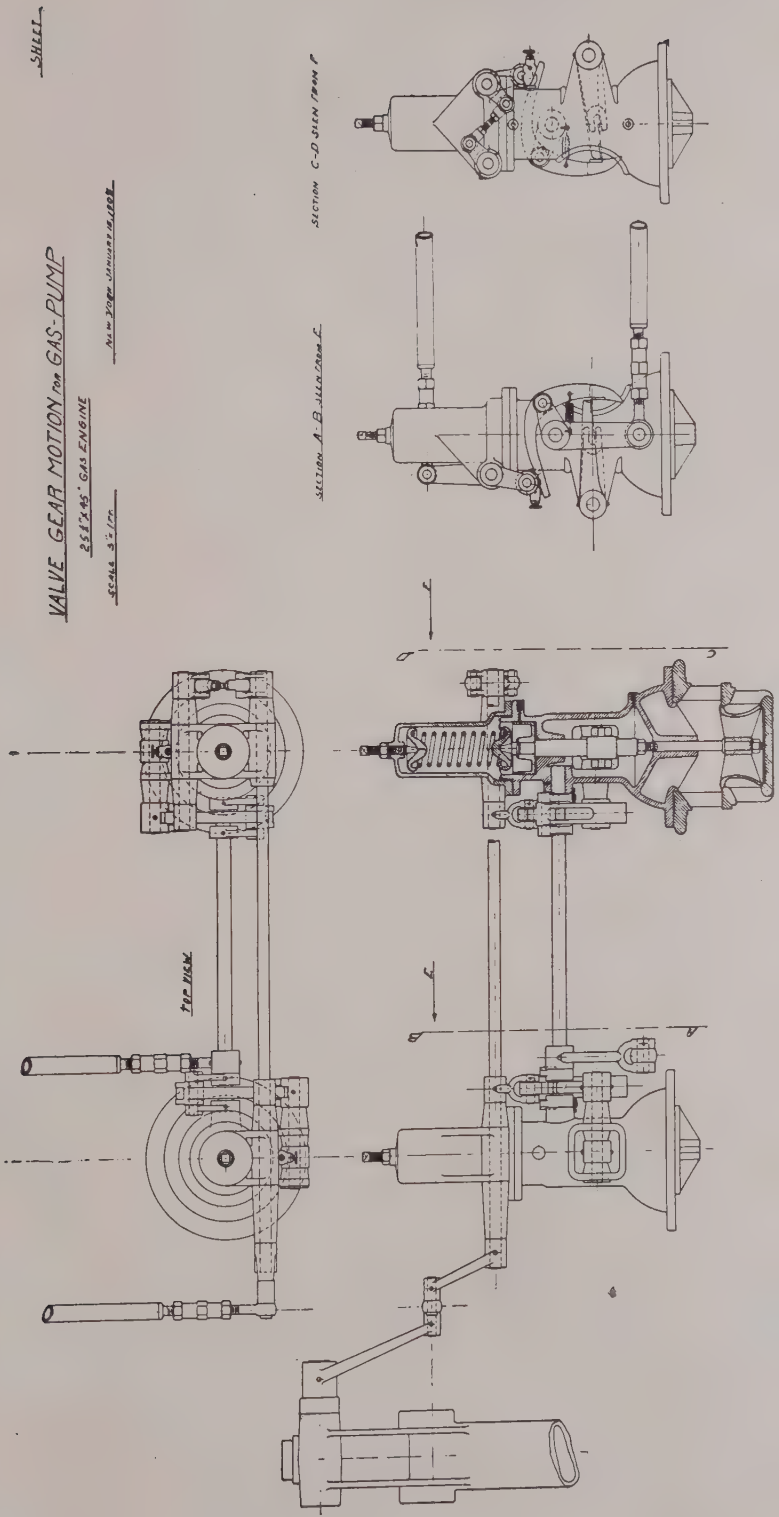

SHEET
VALVE GEAR MOTION FOR GAS-PUMP
25⅝"X45" GAS ENGINE
SCALE 3"=1FT.
NEW YORK JANUARY 18, 1908
SECTION C-D SEEN FROM F
SECTION A-B SEEN FROM E
TOP VIEW
E
F
A
B
C
D

FIG. 579.

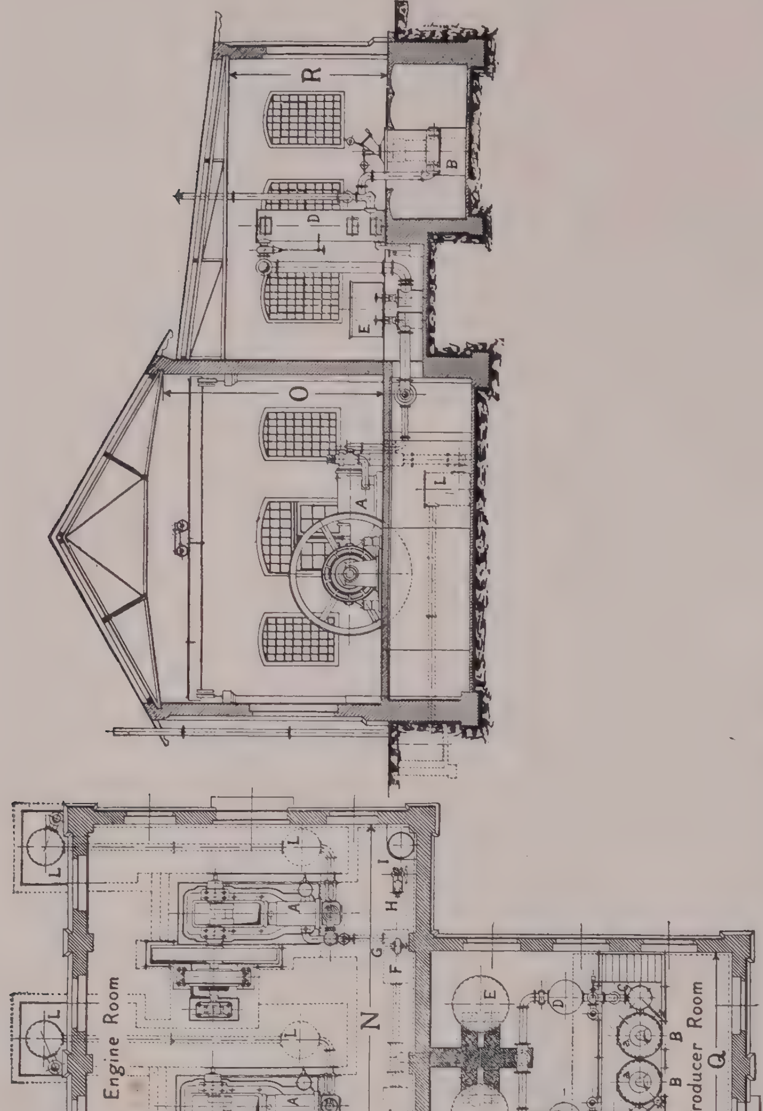

R
B
D
E
O
A
L
L
I
H
G
F
A
L
Engine Room
N
L
A
F
M
Switchboard
Office
E
D
E
D
B
B
C
C
Producer Room
Q
P
J
K
Hall
Workshop
Stores
Coal Bunker

FIG. 580.

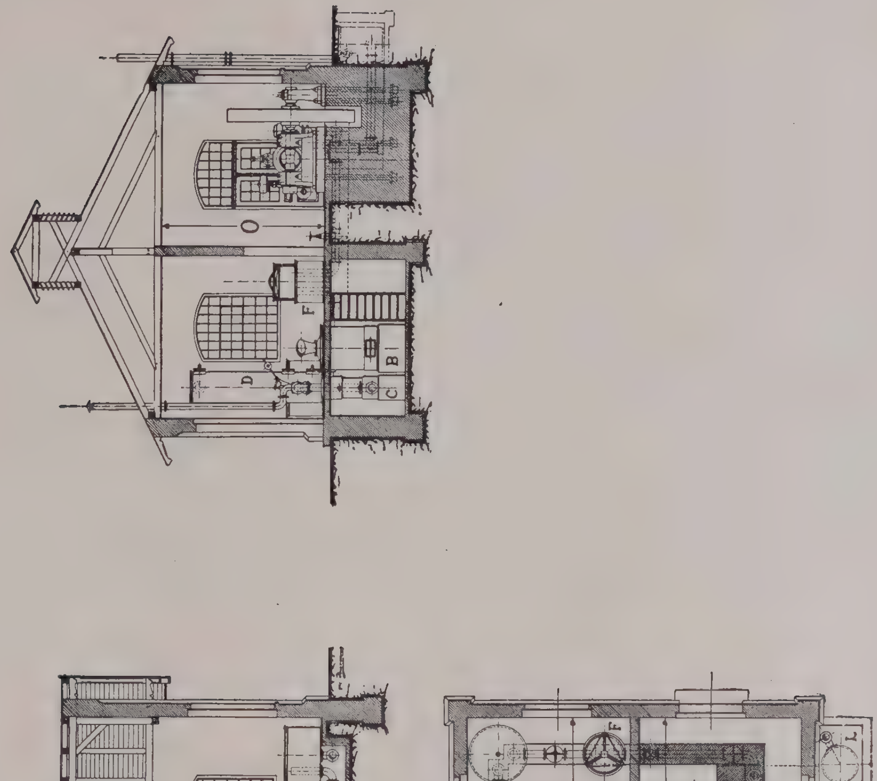

O
F
D
B
C
E
D
A
C
D
E
B
F
I
H
K
G
Q
P
N
M
A
L
Producer Room
Engine Room
Coal Bunker
Switch Board
Electric Storage Batteries

Fig. 581.

3. *The Hornsby-Akroyd Oil Engine.* (Type HA). This is a very popular English 4-cycle oil engine, placed on the American market in 1895. Fig. 582 is a general view of a 20 B.H.P. size of this type. The constructive details are illustrated in Figs. 583, 584 and 585, from which its operation should be clear without much explana-

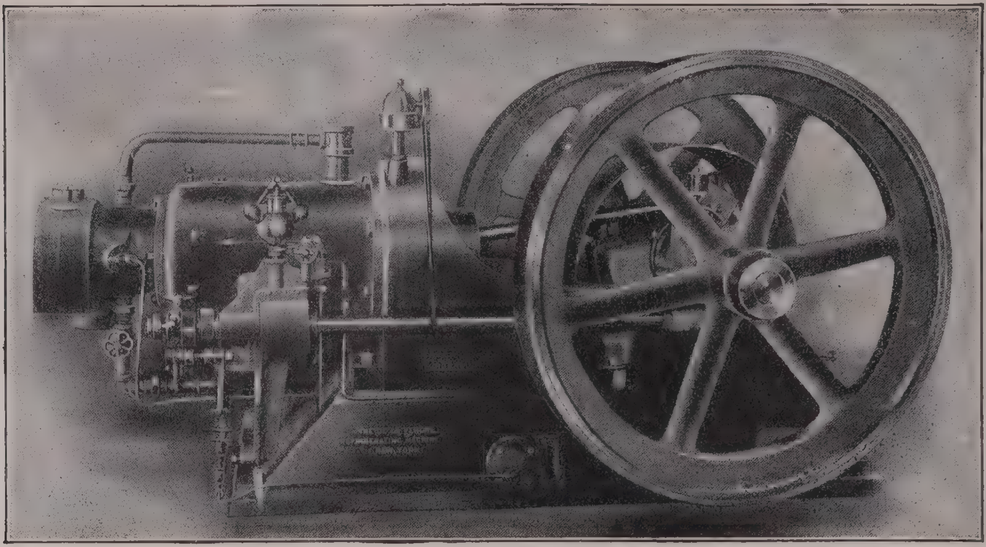


Fig. 582.

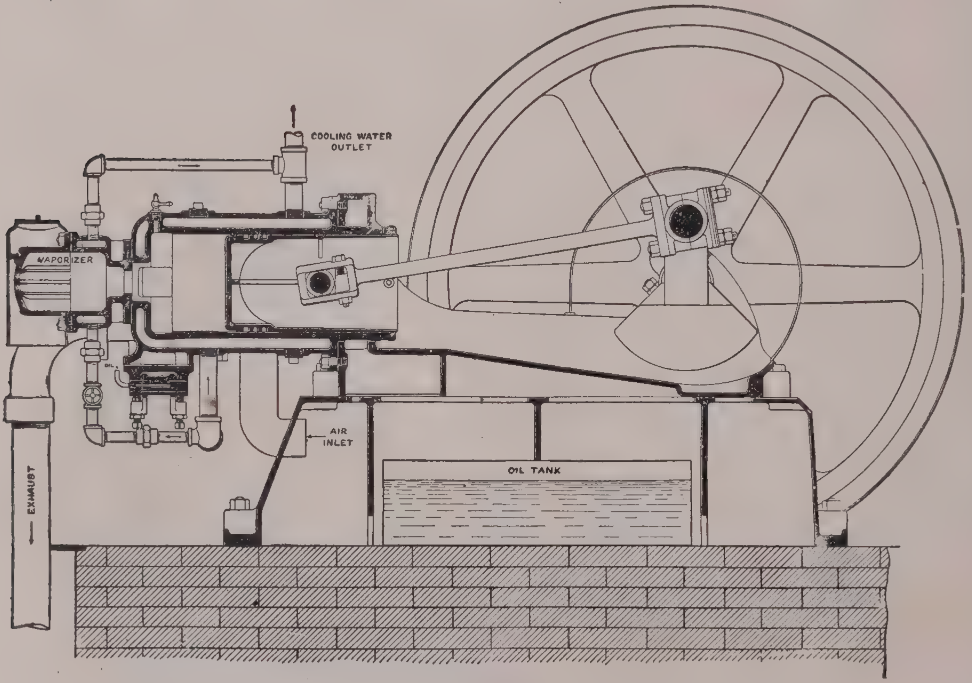


Fig. 583.

tion. On the suction stroke of the piston the oil is injected by a plunger pump, operated by the air valve cam on the lay shaft, through an atomizing plug into the vaporizer chamber which is a cast-iron vessel connected to the cylinder by a narrow neck. The oil vaporizes when coming in contact with the hot walls and the mixture is formed when the piston on the return stroke compresses the air into the vaporizer. The compression is such that the mixture ignites when the piston reaches the dead center, no igniter of any kind being required.

The vaporizer is heated before starting by means of a blow lamp. In order to accomplish this inside of ten minutes the De La Vergne Co. make a special heating lamp which works on the principal of the ordinary gasoline torch, but uses kerosene as fuel. The lamps used for the smaller size of engine are shown in Fig. 586, comprising one or more burners, while Fig. 587 illustrates a special air compressor

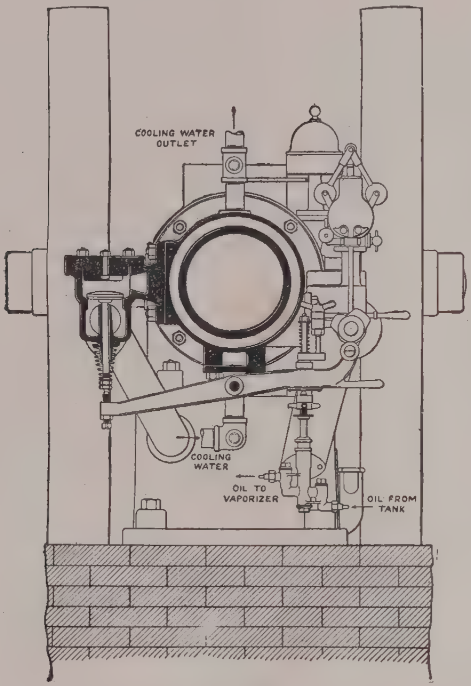


Fig. 584.

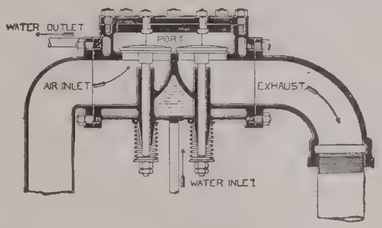


Fig. 585.

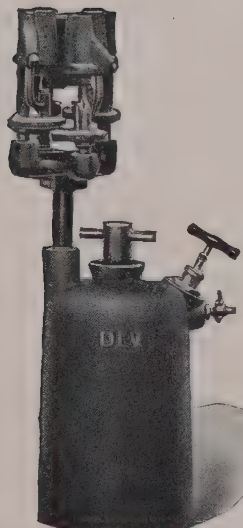


Fig. 586.

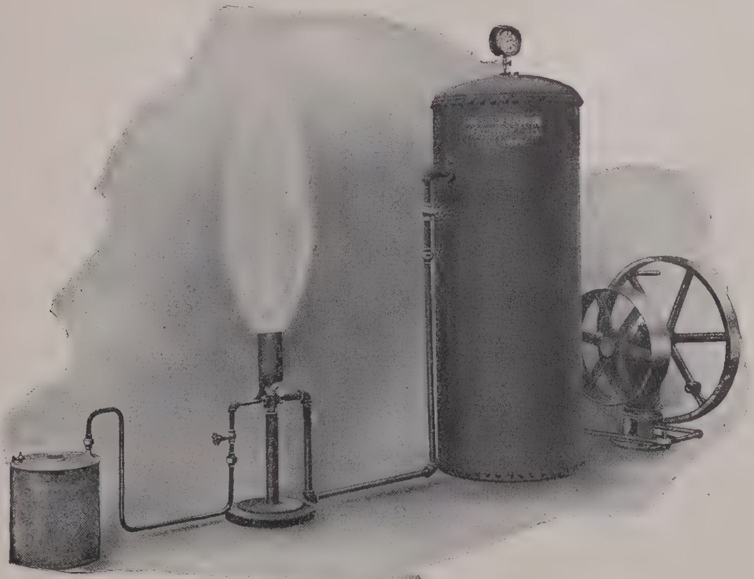


Fig 587.

outfit and lamp used for the larger sizes. The compressor and receiver can also be used for starting the engine.

After starting, the heat of compression and explosion is sufficient to maintain the vaporizer cap at a dull red heat. A protecting hood placed over the cap at starting can be lifted off if the engine tends to get too warm by running continuously at full load. Using moderate pressures it is practically impossible to burn kerosene or crude oil without the formation of some carbon deposit. Recognizing this fact, the oil is in this engine not injected into the cylinder where the deposits might cause trouble, but into a separate vaporizing chamber.

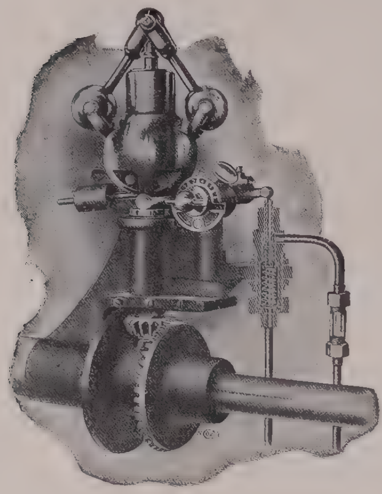
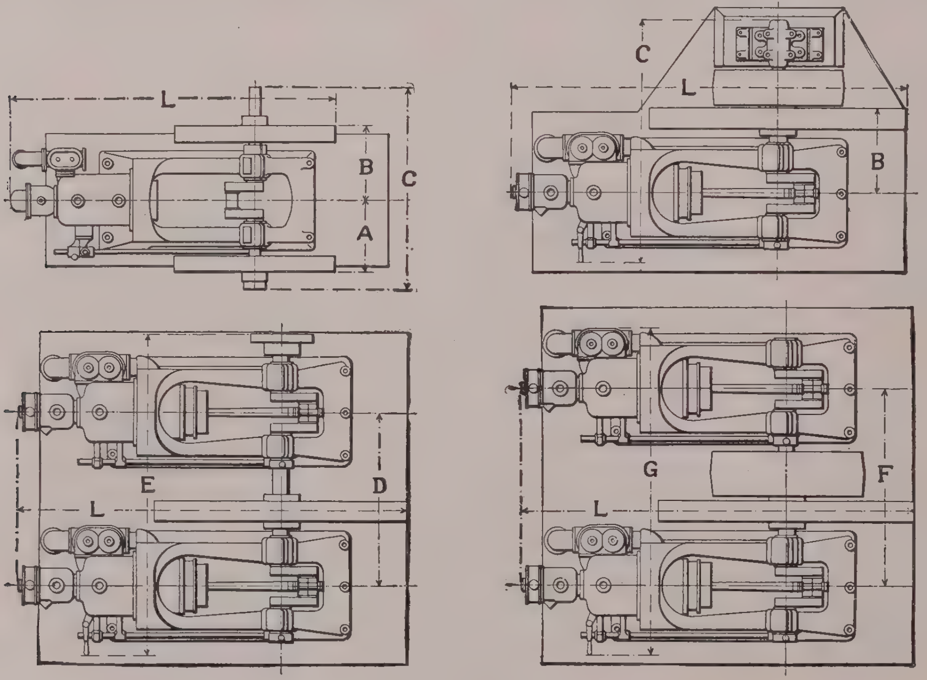


Fig. 588.

Originally the vaporizer consisted of a single gun-iron casting, but about ten years ago it was found that the compression could be increased without incurring pre-ignitions, by simply water-jacketing about one-half of the surface of the vaporizer. In this way the economy was improved and the horse-power increased by 20% without adding to the number of revolutions.

Having the vaporizer in two parts, as indicated in Fig. 583, the removal of the deposits becomes a very simple matter. In the large sizes the hot cap has a hand-hole plate held in place by yoke and set screw. Joints are made by means of an endless copper wire ring recessed in the water-jacketed part of the vaporizer. How often the operation of scraping out the carbon may have to be undertaken depends altogether upon the



Figs. 589–592.

nature of oil used, and can only be determined by trial. At best it will have to be done once a month, at worst, once a day. Unless very excessive, the deposits do not interfere with ordinary operation and by keeping duplicate caps on hand an engine can be operated for at least 23½ hours each day in the worst cases.

The speed of Hornsby-Akroyd engines is regulated by proportioning the quantity of oil to the load. The fly-ball governor is shown in Fig. 582, and in greater detail in Fig. 588. It acts upon a by-pass valve so that the oil discharged by the pump (which is always a constant quantity depending upon the setting of the effective pump stroke) is divided into two parts, one going to the vaporizer, the other returning to the fuel tank in the engine base.

These engines are made as single-cylinder units in sizes from 10 to 125 B.H.P., and as twin-cylinder units in sizes from 40 to 250 B.H.P. Figs. 589, 590, 591 and 592, together with the two tables following, show the principal dimensions.

TABLE 88

PARTICULARS OF HORNSBY-AKROYD OIL ENGINES—(HORIZONTAL)

SINGLE-CYLINDER

Brake H.P.	**10**	**15**	**20**	**25**	**35**	**50**	**85**	**125**
Diameter and stroke	9.5×12	11×15	13.5×16	14.5×17	16×20	18.5×25	23×28	27.5×33
Revolutions per minute	290	260	220	220	220	180	180	145
Number of fly-wheels	2	2	2	2	2	1	1	1
Diameter of fly-wheels. . . . . .ft.	4.5	4.75	5	6	6	9	11	12
Face .in.	6	7	7	7	7.5	10.5	11.5	13
A . . . . .in.	18.25	22.62	25.25	28.25	32	. . . .	. . . .	. . . .
B . . . . .in.	18.25	22.62	26.5	28.25	32	38	44.5	53.5
C	4′ 2.5″	5′ 0.5″	5′ 9.75″	6′ 10.5″	7′ 6.75″	9′ 3″	10′ 4.5″	12′ 7″
L	7′ 9″	9′ 4″	10′	10′ 9″	12′ 3.5″	14′ 6″	17′ 4″	20′ 6″
Floor line to shaft center	29.5″	31.5″	32″	38″	38″	36″	36″	45″
Approx. ship. weight, ...lbs.	5000	7500	10400	13600	16000	24000	41000	70000

TABLE 89

PARTICULARS OF HORNSBY-AKROYD OIL ENGINES—(HORIZONTAL)

TWIN-CYLINDER. WITH ONE FLY-WHEEL

Brake H.P.		**40**	**50**	**70**	**100**	**170**	**250**
Diameter and stroke		13.5×16	14.5×17	16×20	18.5×25	23×28	27.5×33
Revolutions per minute		220	220	220	180	160	145
Diameter of wheel		7′	7′	7′ 10.5″	9′	11′	12′
Face of wheel		8″	8″	9″	10.5″	11.5″	13″
With shaft coupling outside	*D*	4′ 4″	4′ 8.75″	5′ 6″	6′ 6″	7′ 6″	9′
	E	8′ 5″	9′ 3″	10′ 6″	12′ 7′ ′	13′ 3.5″	16′ 9″
With pulley inside	*F*	5′ 6″	6′ 3″	7′	8′	8′·6″	11′
	G	9′ 1.5″	10′ 2″	11′ 5″	13′ 2″	14′ 1″	17′ 10″
	L	11′	11′ 3″	13′ 3″	14′ 6″	17′ 4″	20′ 6″
Floor line to shaft center		32″	32″	32″	36″	36″	45″
Approximate shipping weight . . .lbs.		17200	22000	27000	39000	66000	115000

Hornsby-Akroyd engines have found extended application in this country, there being not less than 5315 B.H.P. in one industrial establishment alone. They have also been extensively used for driving the oil pumps in oil pipe line stations. An example of a modern plant of this kind is that of the Gulf Pipe Line Co. in Texas, the lay-out for which is shown in Fig. 593. The oil used for fuel is taken directly from the line.

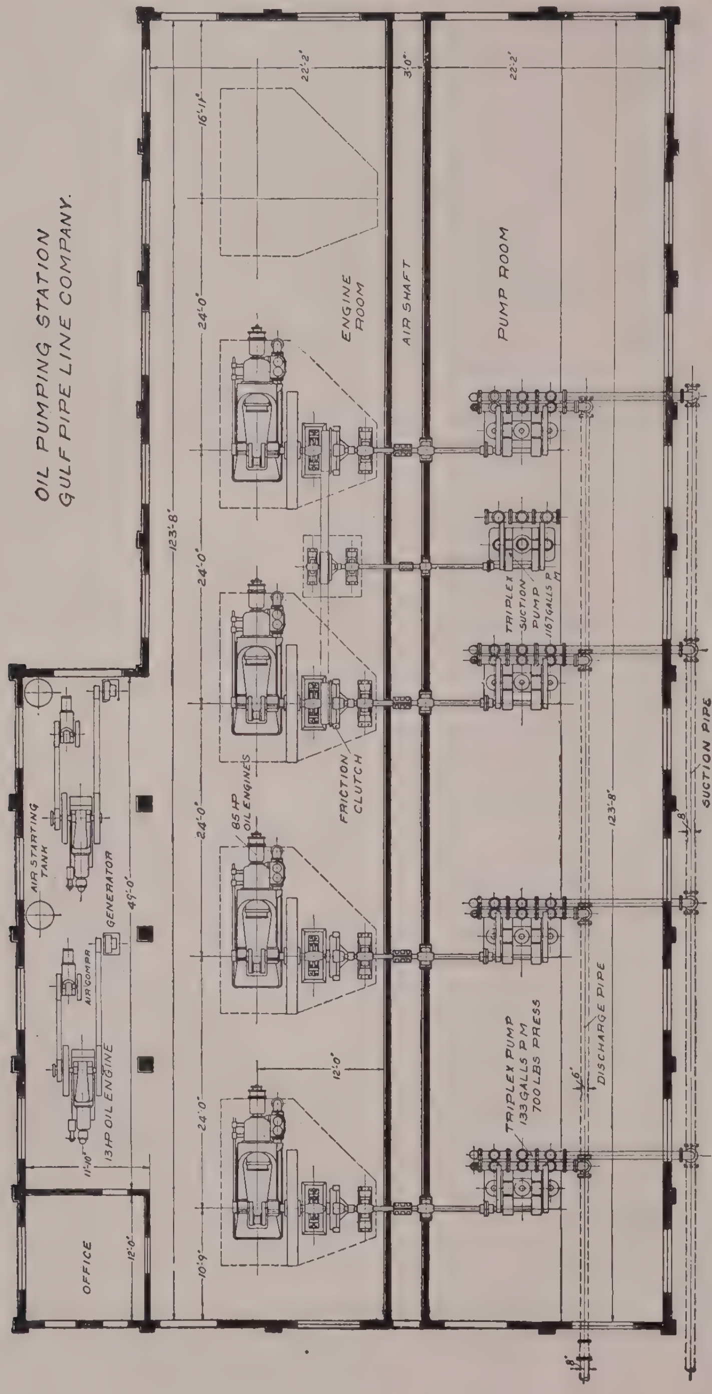

OIL PUMPING STATION
GULF PIPE LINE COMPANY.
22'-2"
3'-0"
22'-2"
16'-11"
24'-0"
24'-0"
24'-0"
24'-0"
123'-8"
ENGINE ROOM
AIR SHAFT
PUMP ROOM
TRIPLEX SUCTION PUMP 1167 GALLS P M
FRICTION CLUTCH
85 HP OIL ENGINES
AIR STARTING TANK
GENERATOR
49'-0"
AIR COMPR
13 HP OIL ENGINE
11'-10"
OFFICE
12'-0"
10'-9"
12'-0"
TRIPLEX PUMP 133 GALLS P M 700 LBS PRESS
DISCHARGE PIPE
6"
8"
123'-8"
SUCTION PIPE
18"

Fig. 593.

There are three stations, covering a total length of about 300 miles. The pressure pumped against is about 700 lbs. per sq.in., each station pumping in the neighborhood of 576 000 gallons per day of 24 hours.

4. *The De La Vergne Vertical 2-cycle Oil Engine.* (Type S.) This is a small engine recently developed. Fig. 594 explains the construction and at the same time shows the appearance of the stationary type, Fig. 595 illustrating the marine type. Both of these illustrations show two-cylinder engines, but single-cylinder machines are also made, as may be seen from the following table of dimensions:

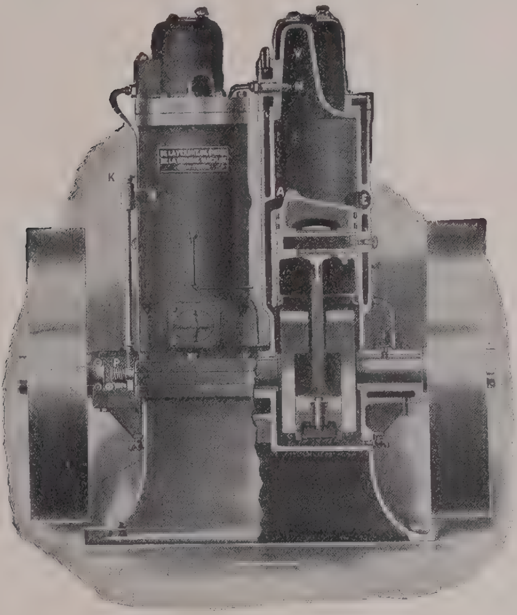


Fig. 594.

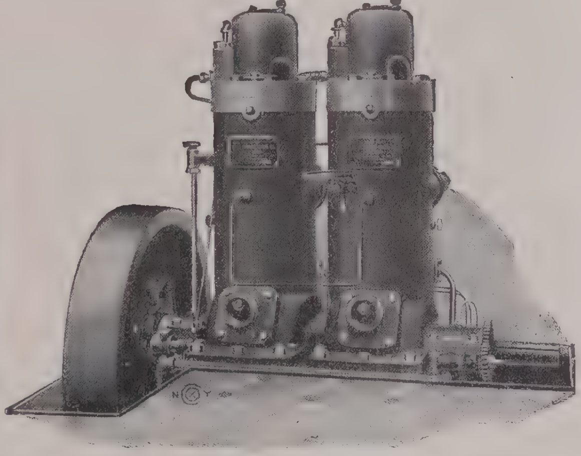


Fig. 595.

Table 90

PARTICULARS OF DE LA VERGNE VERTICAL OIL ENGINES

All Dimensions in Inches

Brake H.P.	No. of Cylinders.	Diameter and Stroke.	R.P.M.	Floor Space (Size of Base).	From Floor or Bottom of Base to Shaft Center.	Height above Shaft Center.	Diameter and Face of Standard Fly-wheel.	Total Shipping Weight. Approx.
				STATIONARY ENGINES				
4	1	5× 5.5	650	14×22	14	23.75	20×4.5	450 lbs.
8	2	5× 5.5	650	23×22	14	23.75	20×4.5	700 “
7½	1	7× 7.5	500	19.5×29.5	16½	32.62	28×5.25	1100 “
15	2	7× 7.5	500	30.5×29.5	16½	32.62	28×5.25	1600 “
12½	1	8×10	450	23 ×38.25	20	43	30×5.25	1800 “
25	2	8×10	450	36.5×38.25	20	43	30×5.25	2700 “
				MARINE ENGINES				
4	1	5× 5.5	650	6×15	5.625	23.75	20×4.5	350 lbs.
8	2	5× 5.5	650	14.75×15	5.625	23.75	20×4.5	550 “
7½	1	7× 7.5	500	10×20	7	32.62	26×5.25	900 “
15	2	7× 7.5	500	21×20	7	32.62	26×5.25	1300 “
12½	1	8×10	450	11.5×26	9.5	43	30×5.25	1400 “
25	2	8×10	450	25×26	9.5	43	30×5.25	2200 “

The engine operates on the ordinary 2-cycle principle, compressing the air for scavenging and charging in the crank case. The vaporizer chamber *V*, Fig. 594, is merely the head or extension of the cylinder and not a separate chamber with a contracted neck, as in the Hornsby. Oil is injected through the plug *N* by means of a plunger pump *P* just as the piston reaches the upper dead center. The oil vaporizes almost instantly as the nozzle *N* projects into the cylinder and is thus kept hot. A cross-section of the spray nozzle is shown in Fig. 596. *K*, Fig. 594, is a plunger arrangement by which oil may be injected by hand into the cylinder on starting. It can also be used for changing the pump stroke.

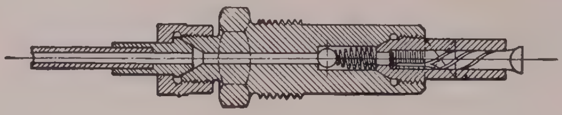


Fig. 596.

For the purpose of keeping the vaporizer hot, these engines, like the Hornsby, are governed by regulating the quantity of oil required per power stroke. The governor is shown in Fig. 597 and its action is described as follows in the catalogue of the company: Frame *F* is fastened with two studs concentrically to the inside of the fly-wheel. To this frame is hinged the cam ring *R* which, on the fly-wheel side, has a projection *B*, and back of that, cam projection *C*.

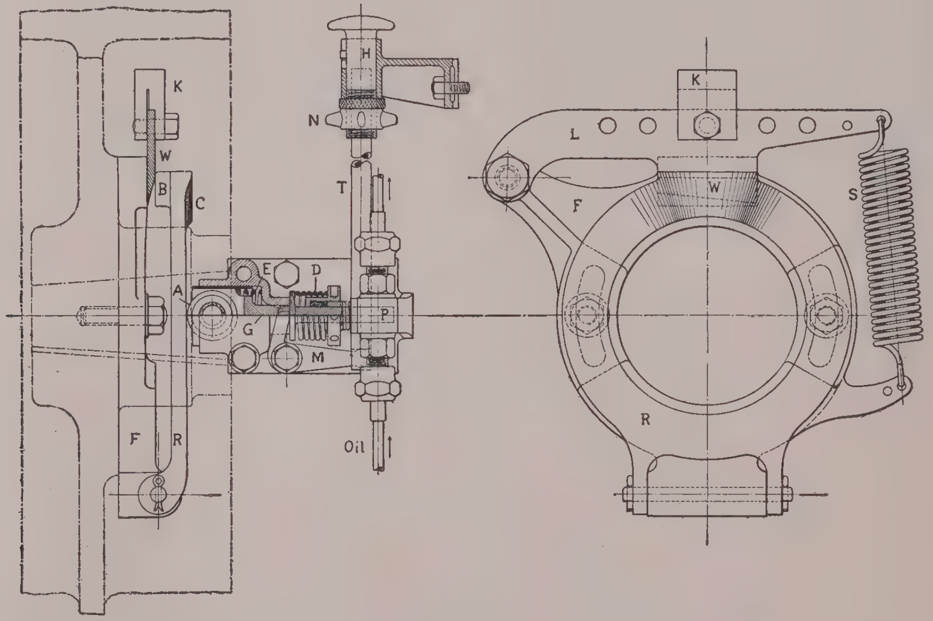


Fig. 597.

Once in each revolution of the fly-wheel, cam *C* pushes roller *A*, thus moving oil pump plunger *P*. The stroke imparted to the plunger is gauged by the lever *L* pivoted to frame *F*. In the middle of lever *L* is a wedge *W*, which more or less separates cam ring *R* from frame *F*.

When the engine is started, the full charge of oil is injected, and the governor does not act till the normal speed is approached. Under light load, the speed tends to accelerate when, by the action of centrifugal tendency, the counterweight *H* will overcome the tension of spring *S* and thus draw the wedge away from the crank-shaft, more or less. Spring *E* always keeps roller *A* riding on cam ring *R*. If, therefore, the wedge is withdrawn, buffer *G* moves away from plunger *P*, and to that extent the effective lift of *C* is reduced, and with it the amount of oil injected. If, at excessive speed, the wedge is entirely withdrawn, spring *E* can push buffer *G* so far away from plunger *P* that, even when pushed back again by *C*, the plunger will not be touched, and no oil will be pumped, compelling the engine to come down to its normal speed.

The two concentric slots in frame *F* allow of making an adjustment when it is desired to greatly change the normal speed of the engine, the effect being that the time of injection, which is coincident with the time of ignition, is advanced or retarded relative to the upper dead point of the crank. For smaller speed variations, the spring *S* and weight *H* can be shifted in the holes of lever *L*. By adjusting the lock nuts *N* of the throttle bar *T*, which limit the stroke of plunger *P* by means of bell crank *M*, the oil supply can also be diminished and thus the speed reduced by about 10%.

Operating Results. *The Körting 2-cycle Engine.* There are apparently no tests available on Körting 2-cycle engines in America. The company bases the rating for these engines upon a mean effective pressure of 65 lbs. for producer and blast furnace gas, and 75 lbs. for natural gas. In order to prevent pre-ignition at the high compression carried, 150 lbs., the charge is diluted with cooled exhaust gas. The mechanical efficiency is from 70–75%, including the pump work. Engines are able to carry about 15% overload. A recent guarantee on a 500 B.H.P. engine running on natural gas was 11000 B.T.U. per B.H.P. hour, which corresponds to an economic efficiency of 23.1%.

The Körting 4-cycle Engine. In these engines, running on producer gas, the compression pressure at full load is from 160 to 180 lbs. gauge, the maximum pressure from 300 to 350 lbs., terminal pressure about 32 lbs., M.E.P. about 65 lbs. The mechanical efficiency varies between 82 and 87%.

(*a*) Test of 75 B.H.P. engine and producer at the Merrimac Chemical Co.'s Works, North Woburn, Mass., made by Professor Miller of the Massachusetts Institute of Technology, for the purchaser, and Mr. A. Lebrecht for the company, June, 1907. Motor cylinder 17×27½″, normal speed 170 r.p.m. Fuel used was coke of the following composition:

Moisture	12.14
Volatile matter	3.05
Fixed carbon	77.58
Ash	7.23
Heating value	13 839 B.T.U. per lb. of dry coke.

The engine was belted to an air compressor and run continuously for 72 hours at full load. Producer was charged regularly every hour, but only twice poked down and cleaned out during the run. The gas made carried considerable tar, and the tar extractors which were installed near the engine were cleaned at intervals of from 5–8 hours without affecting the working of the engine. During such periods about ¼ lb.

of tar would collect on the screens. The gas was analyzed from time to time, the result being shown in the following table:

Sample.		*A*	*B*	*C*	*D*
Date taken June, 1907—Time.		10th, 11.10 A.M.	12th, 3.20 P.M.	13th, 9.30 A.M.	13th, 2.30 P.M.
Carbon dioxide,	CO_2	6.20%	5.60%	6.20%	6.00%
Oxygen,	O	.20	3.00 *	.40	1.20
Carbon monoxide,	CO	26.80	23.60	27.80	26.00
Hydrogen,	H_2	13.47	12.14	13.33	14.40
Methane,	CH_4	0.10	0.10	none	0.20
Nitrogen,	N	53.23	55.56	59.27	52.20
Heat value in B.T.U. per cubic foot	High	142	126	143	143
	Low	134	119	136	135

* Gas sample *B* happened to contain some air.

On a preliminary run, during which the engine was loaded down by a Prony brake, it was shown that for a B.H.P. of 75, the M.E.P. of the cylinder was 64.33 lbs., equivalent to 85.7 I.H.P. This shows a mechanical efficiency of 87.5%. During the main test the load was obtained by an air compressor as above stated, the air being used around the works. Cards taken every 15 minutes during the test showed that the load was fully maintained, as may be seen by examining the cards of Fig. 598 together with the table following.

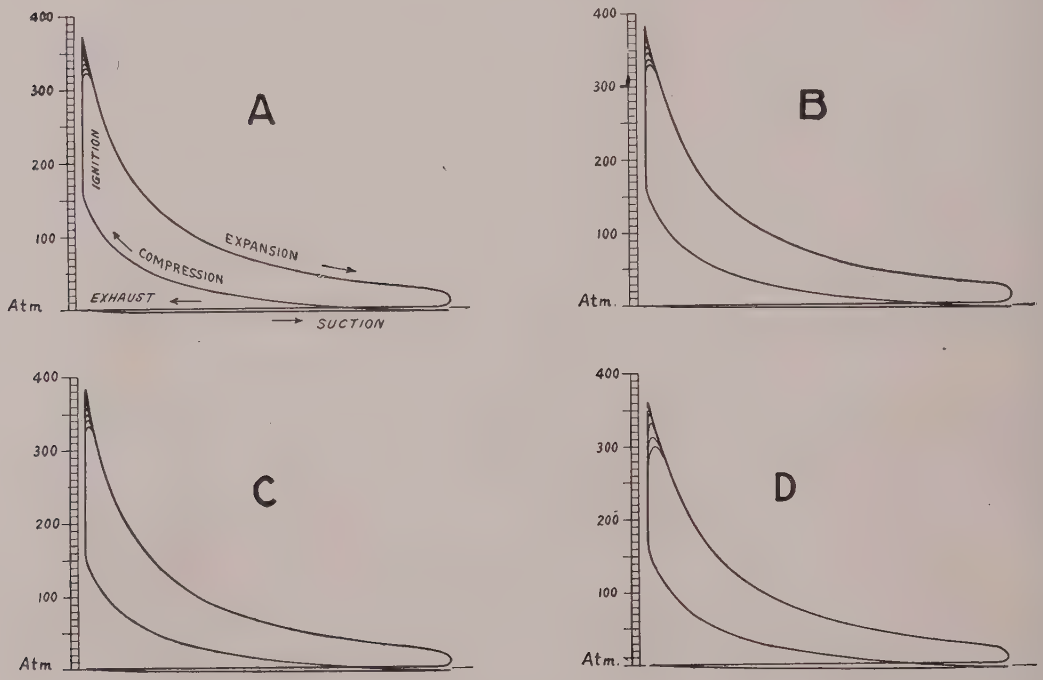


FIG. 598.

Table 91

CARD DATA FOR TEST OF 75 B.H.P. 4-CYCLE ENGINE

Card.	*A*	*B*	*C*	*D*
Date taken, June	10th	11th	12th	13th
Time, P.M.	5.25	4.45	4.40	12.45
Revolutions per minute	172	170	171	170
M.E.P., lbs	64.3	66.7	66.0	65.34
I.H.P.	88.46	88.91	88.50	87.10

The economic results were as follows:

Duration of run, hours	72
B.H.P. developed	75
Total coke used, lbs.	5548
Coke per hour, lbs.	77.06
Coke appearing in ash, %	6.7
Coke per B.H.P. hour on test, lbs.	1.027
Coke per B.H.P. hour, guaranteed, lbs.	1.200
Water evaporated for producer, per B.H.P. hour, lbs.	1.13

This plant possesses a relay unit of the same size. The engines are changed off every 24 hours, although in several instances the shift is not made until 48 hours have elapsed. The producers are changed off about once in three weeks for cleaning purposes.

(*b*) Test of a 200 B.H.P. twin-cylinder gas engine at Petroleum Iron Works, Sharon, Pa., Dec. 28, 1907, by Mr. Kelly, chief engineer for the purchaser, and Mr. A. Lebrecht for the company. Motor cylinders $19\frac{1}{4}\times31\frac{1}{2}''$, normal speed 155 r.p.m. Direct connected to one 125 K.W. D.C. Western Electric generator. The engine will eventually run on producer gas, but natural gas is used as long as the supply lasts.

This test is rendered somewhat unsatisfactory on account of the short time it lasted. The following table shows the principal results, the reading being taken every five minutes:

Table 92

Duration of test, 11:20 to 11:50 A.M., min.	30
Average load, K.W.	143.22
Average load, E.H.P.	192.00
Efficiency of generator, %	92.0
Average B.H.P.	209.0
Average M.E.P., engine No. 78, lbs	70.4
Average M.E.P., engine No. 79, lbs	70.34
Average I.H.P., engine No. 78	125.8
Average I.H.P., engine No. 79	128.0
Total average I.H.P.	253.8
Mechanical efficiency, %	82.4
Gas consumed per hr., engine No. 79, cu.ft.	836
Gas consumed per hr., engine No. 78, cu.ft.	1012
Total gas consumption per hr., cu.ft.	1848

Average composition of gas, vol. %,	CO_2	.20
	O	1.70
	CO	.10
	Illuminants	.40
	Hydrocarbon vapors	3.40
	CH_4	91.80
	H	.23
	N	2.17
Heating value of gas, B.T.U. per cu.ft.,	Higher	1055.5
	Lower	951.5
Gas per B.H.P. hour (corrected for meter), cu.ft.		8.83
Consumption of heat per B.H.P. hr., B.T.U.		8400
Consumption of heat per B.H.P. per hr., B.T.U., guaranteed		9000
Consumption of heat per I.H.P. hr., B.T.U.		6920
Thermal efficiency on I.H.P., %		36.8
Thermal efficiency on B.H.P., % (economic efficiency)		30.3

(*c*) Test of a 150 B.H.P. producer gas engine for Penn Hardware Co., Reading, Pa., Feb., 1907.

Twin-cylinder engine 17×27½″, normal speed 170 r.p.m., but on account of trouble with belt tested at 162 r.p.m. Belt connected to 60-cycle A. C. generator. The gas was furnished by a 300 H.P. "Automatic" pressure producer. The following table shows the main results:

TABLE 93

Duration of test, hours			6
Average B.H.P.			148.5
Average I.H.P.			176.0
Mechanical efficiency, %			84.4
Heating value of coal (higher), B.T.U.			11377
Analysis of coal:	Moisture, %		1.12
	Volatile matter		3.94
	Fixed carbon		76.03
	Ash		18.91
Time of taking gas sample		10 A.M.	2:45 P.M.
Composition of gas, volume, %:	CO_2	3.4	2.4
	O	1.2	.50
	CO	26.0	27.5
	H	7.12	10.27
	CH_4	.70	.90
	N	61.58	58.43
Heat value of gas, B.T.U. per cu.ft.:	Higher	116.6	135.
	Lower	113.1	128.5
Coal consumed per B.H.P. hr., lbs. (including that in boiler)			1.24
Gas per B.H.P. hr. (drop holder test), cu.ft.			73.8
Heat consumed per B.H.P. hr. (in gas at 125.8 B.T.U.)			9500
Heat consumed per B.H.P. hr. (in coal at 11377 B.T.U.)			14107
Thermal efficiency of engine on I.H.P., %			31.6
Thermal efficiency of engine on B.H.P., %			26.8
Thermal efficiency of plant on B.H.P., %			18.0

The last two lines of the table show that the efficiency of the producer was $\frac{18}{26.8}=67.2\%$, which is fair considering that it was operated far below its normal load.

The Hornsby-Akroyd Oil Engine. In these engines the compression pressure is only from 45 to 55 lbs., the maximum pressure from 180 to 200 lbs., the terminal pressure 15 lbs., all by gauge. The M.E.P. at full load is from 40 to 42 lbs.

(*a*) Tests given in catalogue of the De La Vergne Co. from "Gas and Petroleum Engines," by Robinson.

TABLE 94

TESTS OF 25 H.P.

Load.	Maximum.	Full.	⅔	⅓	No Load.
Duration of trial, hours	¼	3	3	2	1
R.p.m., mean	203	202.6	202.4	203	201.5
M.E.P., pounds per square inch	...	45.4, 43.4	31.2	18.3	6
I.H.P.	...	32.3, 31.0	22.4	13.1	4.28
B.H.P.	39	26.74	17.96	9	......
Mechanical efficiency	...	82.4, 86.0	80	69	......
Oil used per I.H.P., hour, lbs.	...	0.61, 0.63	0.74	0.91	1.34
Oil used per B.H.P. hour, lbs.	...	0.74	0.91	1.30	......

(*b*) Test of a 20 B.H.P. engine made by Prof. W. Robinson, Feb., 1905.

Normal speed of engine, 245 r.p.m., motor cylinder 11×17½″. Fuel used was "Russoline" having a specific gravity of 0.825 at 60° F.

TABLE 95

RESULTS OF TRIALS OF HORNSBY OIL ENGINE (1905 TYPE) AT GRANTHAM, FEB. 23, 1905

Power.	Full.	¾	½	Light.
Time taken to start	4 min.	.....	.....	.....
Brake horse-power	21.1	15.95	10.75	0
Indicated horse-power	26.4	.....	.....	6.3
Mechanical efficiency	0.80	.....	.....	.....
OIL CONSUMPTION.				
Total in engine, lbs	13.5	10.5	8.5	5.25
Oil per indicated horse-power hour, lbs.	0.5	.....	.....	0.83
Oil per brake horse-power hour, lbs.	0.64	0.66	0.79	.....
Oil per brake horse-power hour, pints.	0.62	0.64	0.76	.....

No further information is given regarding the fuel oil, but assuming that its heating value is 18 500 B.T.U. per lb., the oil consumption at full load would correspond to an economic efficiency of $\frac{2545}{.64\times 18\,500}=21.5\%$.

(*c*) Fig. 599 shows graphically the results of a fuel consumption test of an 85 B.H.P. Hornsby-Akroyd engine made in Sept., 1906.

The cylinder dimensions were 23×28″, the normal speed 160 r.p.m. In this engine the piston was water-cooled. The fuel used was Standard Oil Co.'s fuel oil, but no further particulars with regard to fuel are given. The best consumption was apparently .84 lb. of oil per B.H.P. hour at 75 B.H.P. Assuming again that the oil has a heating value of 18 500 B.T.U. per pound, this figure would correspond to an economic efficiency of $\frac{2545}{.84\times 18\,500}=16.4\%$.

The De La Vergne 2-cycle Vertical Oil Engine. The following averages at full load are representative of this type: compression 89 to 110 lbs., maximum pressure 250 to 300 lbs., exhaust 50 lbs., M.E.P. 28 to 33 lbs., mechanical efficiency 75 to 80%.

(*a*) Test of a 7×7½″ twin cylinder engine, rated at 15 B.H.P. Engine direct connected to 10 K.W. Western Electric Co. generator on same base plate. Normal speed 500 r.p.m. Fuel used was kerosene. Tests made April 6 and 7, 1908.

TABLE 96

Load Factor.	1¼	$\frac{1}{1}$	¾	½	¼	$\frac{0}{0}$
Length of test, minutes	15	30	30	20	20	30
R.p.m.	500	507	520	528	535	545
Volts	126	120	120	122	120	0
Amperes	97	83.5	63	44	21	0
Kilowatts	12.22	10.02	7.56	5.37	2.52	0
Electrical H.P.	16.37	13.42	10.13	7.19	3.38	0
Developed H.P.	19.5	15.8	12.08	8.88	4.5	0
Generator efficiency, %	84	85	84	81	75	0
Total oil used	3 lbs. 15 oz.	5 lbs. 4 oz.	4 lbs. 9 oz.	2 lbs. 8 oz.	1 lb. 13 oz.	1 lb. 15 oz.
Oil per hour, lbs.	15.75	10.5	9.12	7.5	5.44	3.88
Oil per K.W. hour, lbs.	1.3	1.05	1.2	1.4	2.16	0
Oil per E.H.P. hour, lbs	0.96	0.78	0.9	1.04	1.6	0
Oil per delivered H.P., lbs	0.81	0.66	0.76	0.86	1.21	0

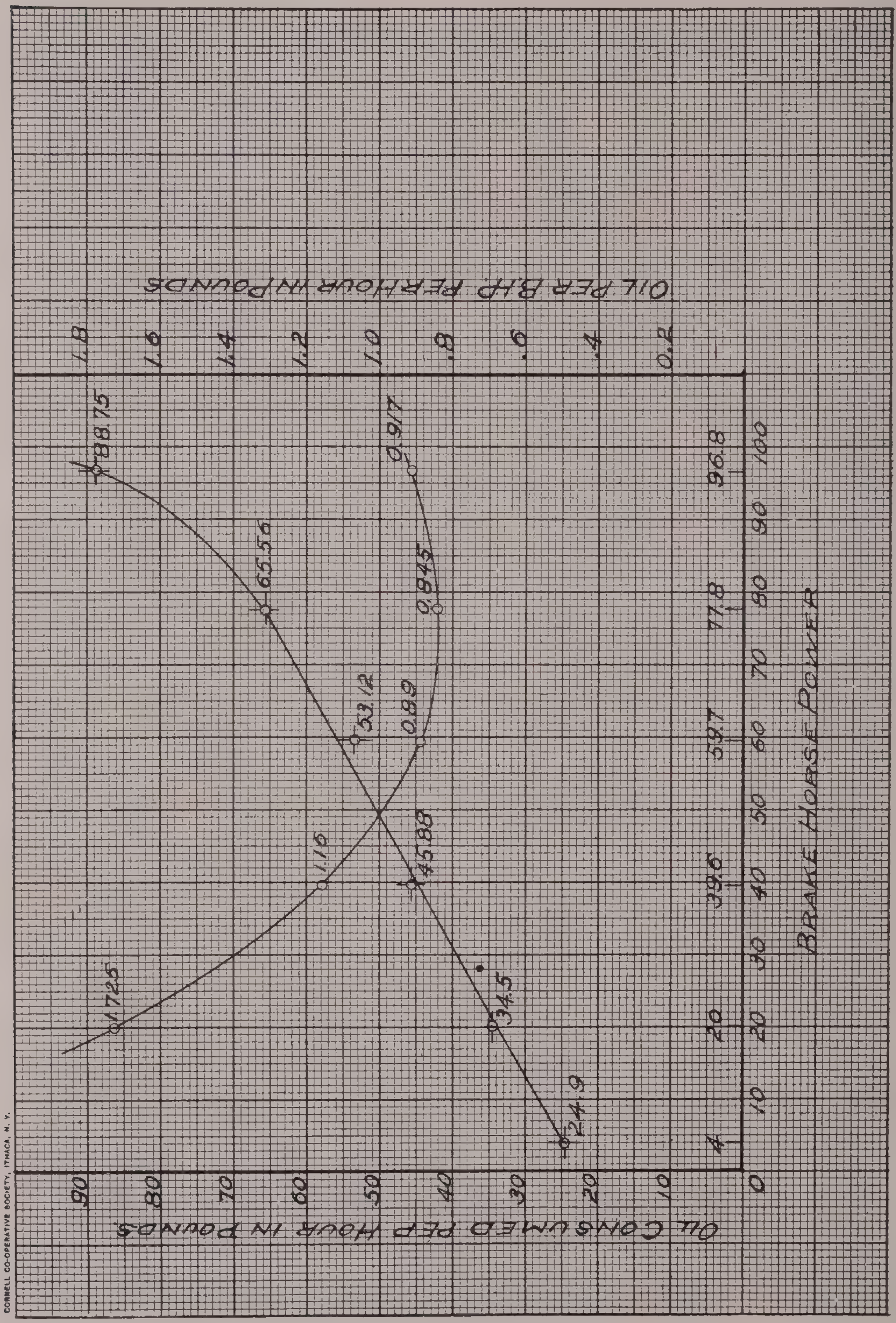

OIL PER B.H.P. PER HOUR IN POUNDS
1.8
1.6
1.4
1.2
1.0
.8
.6
.4
0.2
OIL CONSUMED PER HOUR IN POUNDS
90
80
70
60
50
40
30
20
10
0
BRAKE HORSE POWER
10
20
30
40
50
60
70
80
90
100
4
20
39.6
59.7
77.8
96.8
24.9
34.5
45.88
53.12
65.56
88.75
1.725
1.16
0.89
0.845
0.917
CORNELL CO-OPERATIVE SOCIETY, ITHACA, N. Y.

FIG. 599.

The best consumption is 0.66 lbs. of oil per B.H.P. hour, which, with the assumption above made regarding the heating value of the oil, indicates an economic efficiency of $\frac{2545}{.66 \times 18500} = 20.8\%$. This figure, considering the high speed and small size of the engine, must be pronounced very good.

3. The American Diesel Engine Co., New York City.[1] The Diesel engine, as made by the *Vereinigte Maschinenfabrik Augsburg*, represents the German type and the method of operation, together with some results obtained, have already been discussed on pp. 362 to 366, to which the reader is referred. This engine is to-day made by licensed manufacturers in nearly every country in Europe, and by the above named firm in this country. In all cases the engines built by the various manufacturers of course operate upon the principle already described, but the mechanical details of the various constructions differ here and there enough to make them distinct from each other. Thus the main point in which the American Diesel engine differs from the European types is in the use of the enclosed box frame. The others, to the writer's knowledge, all use the open A-frame.[2] There are also the minor differences in the methods of obtaining compressed air, governor details, valve construction, etc.

Plate XXXI shows the general assembly of a triple cylinder Diesel engine as built by the American Company. The base of the machine is a single casting completely enclosed, so that splash lubrication may be employed for crank-shaft bearings, connecting rod bearings, cam shaft and cam rollers. The cylinders are cast in one piece with the jackets, the whole being secured to the box pedestal as shown. The cylinder head is a simple casting, containing a relief valve in the center of the head, while admission, exhaust, and fuel valves are located in a chamber at the side. Both admission and exhaust valves are of the simple poppet type. The admission valve works downward and the exhaust valve upward. The latter is not water-cooled. All of the valves are operated by push rods, the exhaust valve directly, the admission valve through a lever pivoted to the admission valve housing and the fuel admission valve, which acts horizontally, by means of a bell-crank. All of this will be clear from a study of plate XXXI.

Fig. 600 shows the valve construction in greater detail. The admission valve is held in a separate cage so that the seat may be easily taken care of. It closes against a small dash-pot located in the top of the housing for the purpose of reducing both noise and shock. The exhaust valve has no removable seat, but it is of such dimensions that it may itself be easily removed through the admission valve cage opening, giving opportunity for grinding the seat whenever necessary. The relief valve in the center of the head is usually set to open at about 800 lbs. per square inch and acts as a safety against undue pressure caused by premature ignition, etc. The former may occur when the fuel valve has accidentally stuck, admitting oil to the cylinder during the compression stroke.

The fuel valve, through which oil is admitted to the cylinder after the charge of air is compressed, is a very important part of the engine, and its construction is shown on a still larger scale in Fig. 601. Plate XXXI shows the location of the oil pump at the right hand side of the crank case and the manner of operating it by means of spur gearing from the cam shaft. From this pump an oil supply pipe leads to the oil valve shown at the left of section *A–B*, Fig. 601. From here the oil finds its way into the atomizer in the interior of the bushing held in the cast iron fuel valve cage. Both air and oil connections are screwed into this steel bushing so that the valve cage does not have to stand high pressure. The fuel admission valve itself consists of a nickel steel needle which carries a cast iron spring case on its outer end. Normally the spring forces this needle against its conical seat, but as the fuel-valve cam commences to operate, the bell crank shown pushes the needle to the right against the spring and opens the valve. This happens about the time that the main piston reaches the upper dead center on its compression stroke, and highly compressed air from a storage tank then rushes through the atomizer and forces the oil out into the compressed cylinder charge. In order to keep the needle cool and to keep the oil from carbonizing in the atomizer, water is circulated in the space between the steel bushing and the walls of the valve cage, both inlet and outlet pipes being shown in Fig. 601.

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[1] Since the writing of this article, the American Diesel Engine Co. has been bought by Mr. Adolphus Bush. The main offices are now at St. Louis.

[2] See an article by R. Diesel, in the Zeitschrift d. V. D. Ing., 1903, p. 1366.

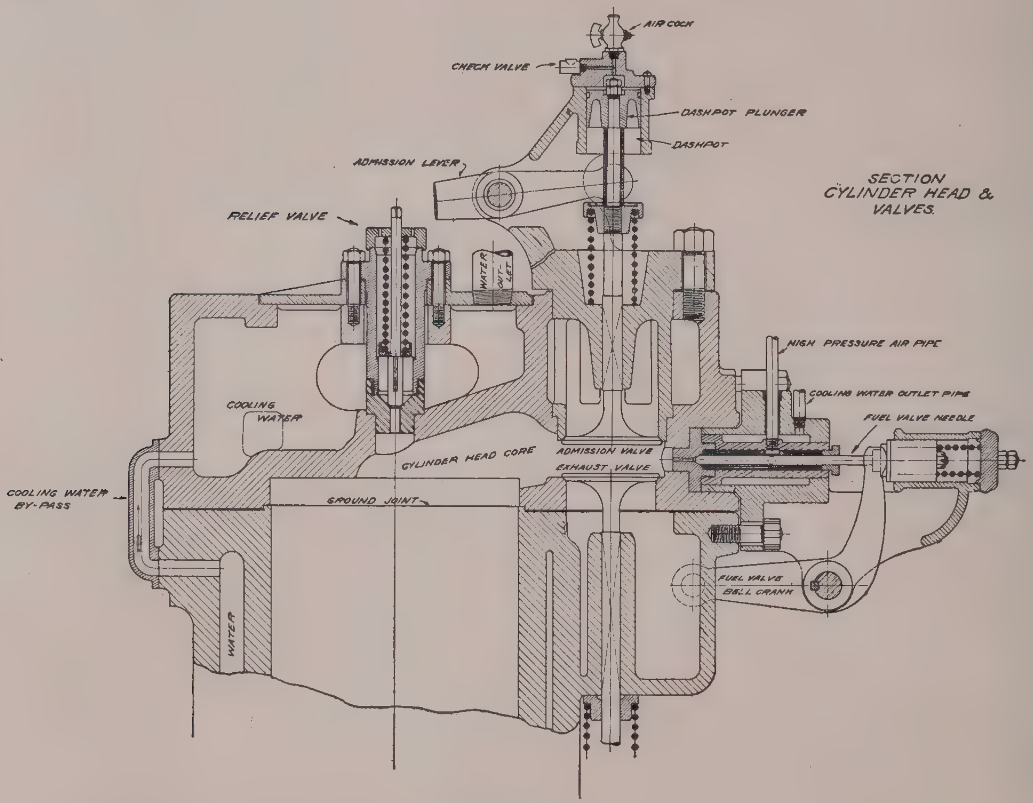


Fig. 600

Diesel engines are governed not by controlling the length of time that the fuel valve is open, but by adjusting the effective delivery stroke of the oil pump, and hence, except for minor variations in the setting of the fuel valve depending upon the kind of oil used, the lift and time of opening of all of the valves once set is always the same. Fig. 602 shows the method of setting the valves and the time of opening and closing, the lift being controlled by the fixed cams of the half-time shaft.

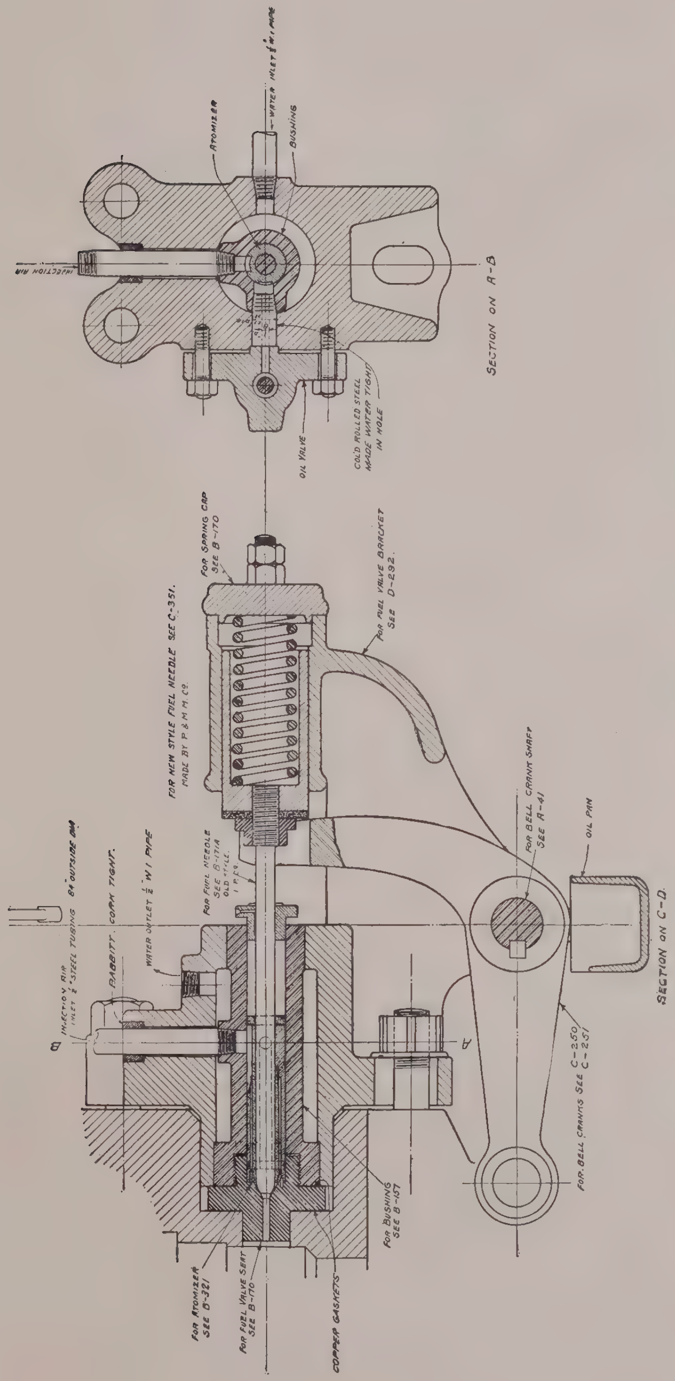
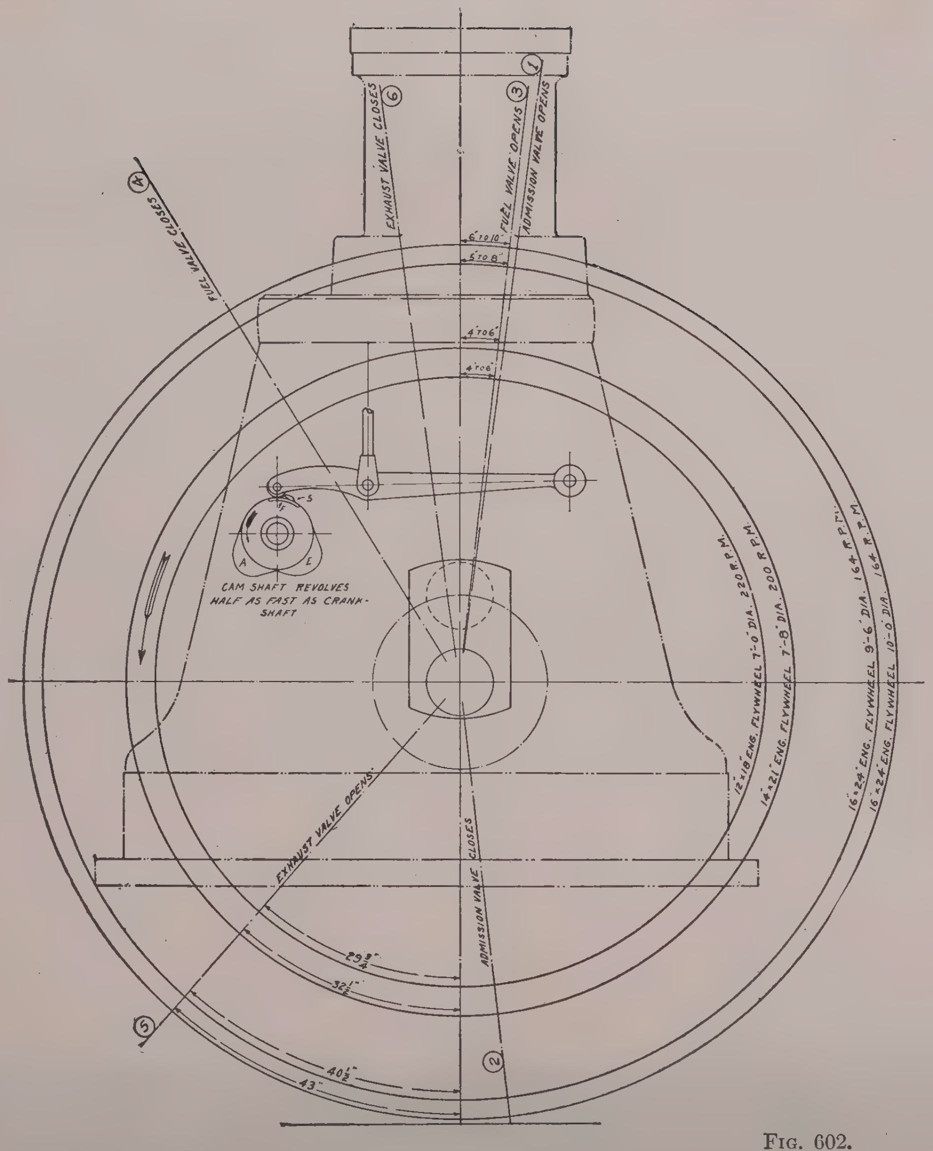
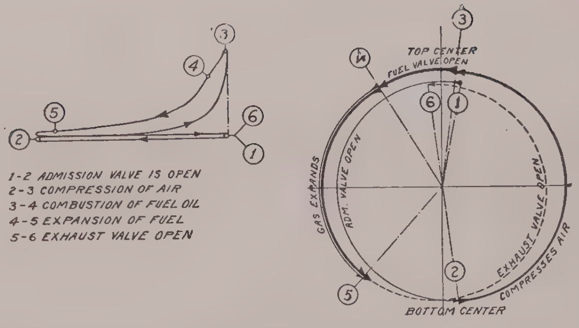

WATER INLET ⅛" W.I. PIPE
ATOMIZER
BUSHING
INJECTION AIR
SECTION ON A-B
OIL VALVE
COLD ROLLED STEEL MADE WATER TIGHT IN HOLE
FOR SPRING CAP SEE B-170
FOR NEW STYLE FUEL NEEDLE SEE C-351. MADE BY P & M M. CO.
FOR FUEL VALVE BRACKET SEE D-292.
INJECTION AIR INLET ⅜" STEEL TUBING ⅝" OUTSIDE DIA.
BABBITT. CORK TIGHT.
WATER OUTLET ½" W I PIPE
FOR FUEL NEEDLE SEE B-171A OLD STYLE
FOR BELL CRANK SHAFT SEE A-41
OIL PAN
SECTION ON C-D
B
A
FOR BELL CRANKS SEE C-250 C-251
FOR BUSHING SEE B-157
COPPER GASKETS
FOR FUEL VALVE SEAT SEE B-170
FOR ATOMIZER SEE B-321

FIG. 601.


EXHAUST VALVE CLOSES ⑥
FUEL VALVE OPENS ③
ADMISSION VALVE OPENS ①
FUEL VALVE CLOSES ④
6" TO 10"
5" TO 8"
4" TO 6"
4" TO 6"
CAM SHAFT REVOLVES HALF AS FAST AS CRANK-SHAFT
A
E
F
S
12"×18" ENG. FLYWHEEL 7'-0" DIA. 220 R.P.M.
14"×21" ENG. FLYWHEEL 7'-8" DIA. 200 R.P.M.
16"×24" ENG. FLYWHEEL 9'-6" DIA. 164 R.P.M.
16"×24" ENG. FLYWHEEL 10'-0" DIA. 164 R.P.M.
EXHAUST VALVE OPENS ⑤
ADMISSION VALVE CLOSES ②
29¾"
32½"
40½"
43"



SIZE OF ENGINE	DIAM. OF FLYWHEEL	MEASUREMENTS ON CIRCUMFERENCE OF FLYWHEEL IN INCHES.						REMARKS
		ADMISSION VALVE		FUEL VALVE		EXHAUST VALVE		
		OPENS BEFORE TOP CENTER	CLOSES PAST BOTTOM CENTER	OPENS BEFORE TOP CENTER	CLOSES PAST TOP CENTER	OPENS BEFORE BOTTOM CENTER	CLOSES PAST TOP CENTER	
12×18	84	5½	4½	4 TO 6	21 TO 23	29¾	5½	
14×21	92	6	5	4 TO 6	23 TO 25	32½	6	
16×24	114	7½	6¼	5 TO 8	28 TO 31	40½	7½	SEE NOTE Ⓕ
16×24	120	7¾	6¾	6 TO 10	30 TO 33	43	8	

NOTE Ⓕ (16×24 ENG) DIFFERENT ADJUSTMENTS OF FUEL VALVE TO SUIT DIFFERENT VARIETIES OF OIL: FOR INSTANCE A TEXAS OIL OF 20° GRAVITY, 8 INCHES WOULD BE REQUIRED TO GIVE THE BEST RESULTS WHILE WITH A PENN'A OIL OF 30° G. 5 IN. WOULD BE SUFFICIENT.

ATOMIZERS SHOULD BE KEPT CLEAR OF SEDIMENT OR GRIT BY TAKING CARE OF THE OIL STRAINERS

FUEL NEEDLE VALVE SHOULD BE CAREFULLY PACKED. IF THE PACKING IS TO TIGHT IT MAY CAUSE THE VALVE TO CLOSE SLOWLY AND THE NEEDLE POINT AND OTHER PARTS ARE LIABLE TO DAMAGE.

FUEL VALVE SETTING: OPEN THE ADMISSION VALVE WITH EASING GEAR & TURN AIR ON IN INJECTION AIR PIPE & SET VALVE ROD SO THE AIR IS HEARD TO ESCAPE THROUGH CYLINDER AT THE TIME GIVEN FOR OPENING OF VALVE.

Fig. 602.

The fuel pump with its governor is shown in Fig. 603. This shows a cross-section through one of the pump cylinders, there being as many pump cylinders as there are power cylinders. The pump plunger is actuated by a simple eccentric and strap, and its stroke is therefore constant. By means of a short horizontal arm and nearly vertical rod, the plunger is connected to a "pump suction valve eccentric lever" which, in conjunction with the fly-ball governor, controls the motion of the suction valve. At full load the governor sets the fulcrum about which the eccentric lever turns so that the suction valve opens when the plunger has completed half of its down stroke. The suction opening increases until the plunger has reached the lower end of its stroke. On the up stroke the valve is not closed until again half the stroke is completed, after which that part of the charge remaining is then forced through the double ball check valve and into the engine cylinder. The reason for keeping the valve open for half stroke each way is to let the oil free itself from air which it is very apt to entrain, and thus to deliver solid oil only. Under no load the governor so changes the motion of the eccentric lever that the suction valve is open practically during the entire up and down stroke of the plunger, so that little or no oil is delivered to the fuel injection valves. Between these two extremes of course the effective delivery stroke of the pump can be made anything to suit the load.

The high pressure air used for injecting the fuel is usually obtained by independent compressors discharging into steel storage tanks. The air for starting is also obtained from this source. The pressure of the injection air should vary with the load on the engine, for half load or less it should be from 50–60 atmospheres, above that load from 65–70 atmospheres and for overloads 75 atmospheres is safe and permissible.

The use of independent three-stage compressors in place of two-stage compressors driven from the engine is considered a distinct improvement over European practice. In a two-unit plant for instance, two independent compressors, one used as a relay, offer much greater security against a shut-down from lack of air than two engine driven compressors. This advantage is more pronounced as the number of units in the plant grows. Thus in a plant in Florida, three compressors serve to supply 12 engine units, and the failure of any one of the three can not possibly effect the operation of the engines. Another important point is that such a system allows of a cool air supply to the injection valves, which helps to prevent the carbonizing of the fuel oil in this valve.

To start the engine the fuel-valve cam on one of the cylinders is pulled over so that the fuel valve on that cylinder is closed and can not be opened. Instead of this the same operation brings into action a cam which controls a special starting valve on the same cylinder. The engine, after the crank-shaft has been brought into proper position, is then started by admitting compressed air to that cylinder. After one of the other cylinders is heard to obtain an ignition, the cam lever is returned to its former position, when the starting cylinder will also take up its regular cycle. Previous to starting the fuel pump must be operated by hand for a few turns by means of the starting wrench and pinion shown in Fig. 603, the pinion meshing with a gear on the pump shaft when the starting pin is pulled up (see the drawing of the cam shaft on Plate XXXI). The oil so pumped is discharged through an overflow, the purpose of this being to work all of the air out of the oil and to insure that nothing but solid oil is delivered to the fuel valves.

The following table shows some details of Diesel engines at present regularly constructed by the American company:

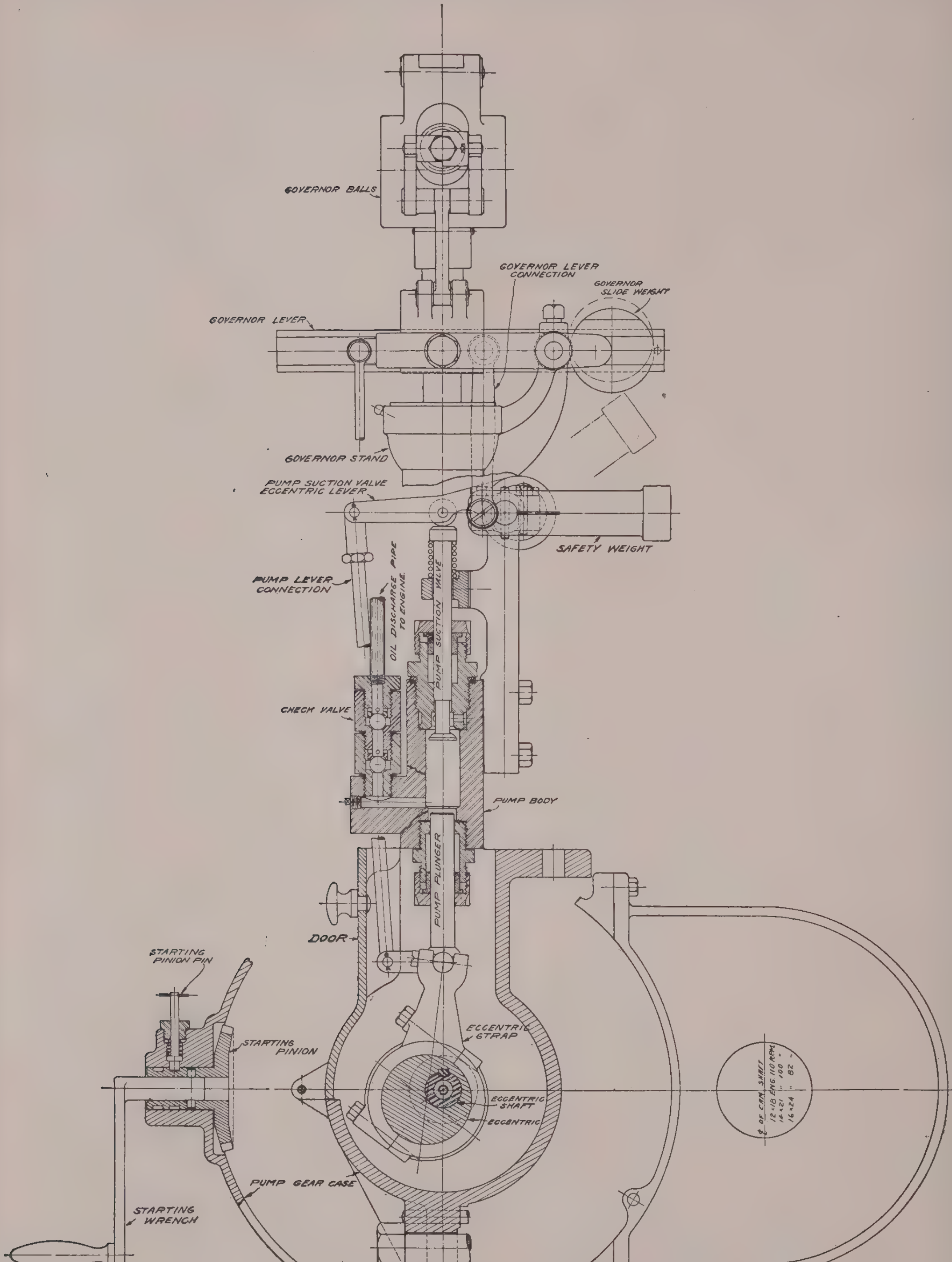

GOVERNOR BALLS
GOVERNOR LEVER CONNECTION
GOVERNOR SLIDE WEIGHT
GOVERNOR LEVER
GOVERNOR STAND
PUMP SUCTION VALVE ECCENTRIC LEVER
SAFETY WEIGHT
PUMP LEVER CONNECTION
OIL DISCHARGE PIPE TO ENGINE
PUMP SUCTION VALVE
CHECK VALVE
PUMP BODY
PUMP PLUNGER
DOOR
STARTING PINION PIN
STARTING PINION
ECCENTRIC STRAP
ECCENTRIC SHAFT
ECCENTRIC
PUMP GEAR CASE
STARTING WRENCH
℄ OF CAM SHAFT
12×18 ENG. 110 R.P.M.
14×21 " 100 "
16×24 " 82 "

FIG. 603.

Table 97

ALL WEIGHTS ARE IN POUNDS

No. of Cylinders.	Cylinder Size.	R.p.m.	Rated H.P.	Weight without Fly-wheels.	Number of Fly-wheels.	Weight of each Fly-wheel.	Total Weight of Fly-wheels.	Total Weight of Engine.
3	10½″×15″	240	75	18500	2	2700	5400	23900
3	12″ ×18″	220	120	27000	2	5900	11800	38800
3	14″ ×21″	200	170	55000	2	6900	13800	68800
3	16″ ×24″	164	225	65000	1	15640	15640	80640
1	16″ ×24″	164	75	28600	1	11800	11800	40400

A lay-out of two 225 H.P. three-cylinder engines driving a 300 K.W. generator, giving some further information regarding dimensions, is given in Fig. 604. In some installations of this type the rotor of the generator is made to act as a fly-wheel, which tends to shorten the length of the plant to a small extent.

Finally, Plate XXXII gives a complete layout and piping plan for three 150 K.W. engines.

Operating Results. (For results with Diesel engines of other makes, see p. 363.)

The number of tests available on American Diesel engines is quite large, and consists partly of acceptance tests made usually by the company's engineers, of operating reports received from a number of plants in commercial operation and a few tests made by independent observers. Except in one or two instances, nothing is stated concerning the heating value of the oil used, although the rest of the information is quite complete. Where efficiency computations are made, the writer has assumed the heating value of the oil at 19 000 B.T.U. per lb., except where otherwise stated. This is in most cases not far from truth, and in any case it is intended to show merely about what Diesel engines are doing as compared with other internal combustion engines.

(*a*) Acceptance Test, Nov. 8–13, 1907, Perth Amboy. Triple-cylinder engine, 16×24″, 164 r.p.m., rated B.H.P., 225. Engine had two fly-wheels, one 8′ 6″×10″, and one 7′ 8″×8″. Load obtained by brake on each wheel. Kind of oil, distillate.

Load.	Duration, Hours.	R.p.m.	B.H.P., Net.	Oil per B.H.P. Hour, Lbs.	Oil per 100 B.H.P. Hours, Gallons.	Economic Efficiency, Per Cent.
⅓	2	173.5	76.1	.60	8.4	22.3
⅔	2	168.6	151.4	.47	6.6	28.5
Full	2	165.8	227.7	.45	6.3	29.8
Full	10	164.8	226.3	.44	6.2	30.5
Over	4	161.9	257.2	.43	6.0	31.3

The engine did not drive its own air compressor. The allowances made at the different loads for this were as follows:

⅓ load	17.6 H.P.
⅔ load	18.4
Full load	19.3
Overload	19.7

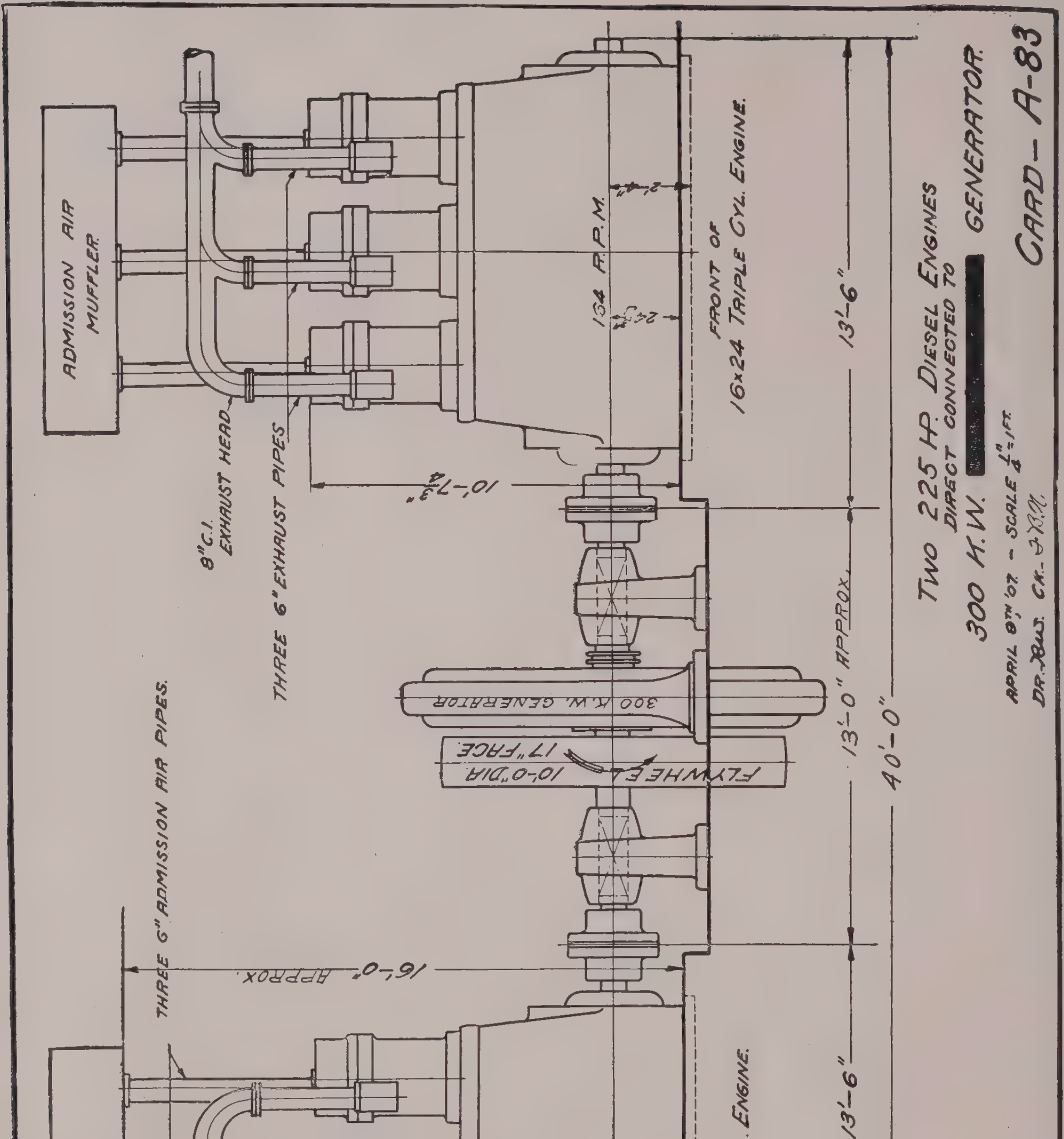

ADMISSION AIR MUFFLER
8" C.I. EXHAUST HEAD
THREE 6" EXHAUST PIPES
10'-7¾"
134 R.P.M.
FRONT OF 16×24 TRIPLE CYL. ENGINE.
13'-6"
THREE 6" ADMISSION AIR PIPES.
300 K.W. GENERATOR
FLYWHEEL 10'-0" DIA. 17" FACE
13'-0" APPROX.
40'-0"
16'-0" APPROX.
ADMISSION AIR MUFFLER
FLOOR LINE
FRONT OF 16×24 TRIPLE CYL. ENGINE.
13'-6"
8" C.I. EXH PIPE
13'-9"
TWO 225 H.P. DIESEL ENGINES DIRECT CONNECTED TO [redacted]
300 K.W. GENERATOR.
APRIL 8TH '07 - SCALE ¼"=1FT.
DR. CK.
CARD - A-83

Fig. 604.

(*b*) Acceptance test, Feb. 9–13, 1906, Bellefontaine. Triple-cylinder engine, 16×24″, 164 r.p.m., rated B.H.P., 225. Load on engine obtained by generator belted to countershaft. Engine drove air compressor belted to countershaft. Kind of oil, distillate.

Load.	Duration. Hours.	B.H.P.	R.p.m.	Oil, Pounds per B.H.P. Hour.	Oil, Gallons per 100 B.H.P. Hours.	Efficiency of Generator and Shaft.	Economic Efficiency, Per Cent.
⅓	2	90.5	165.3	.550	7.58	74.0	24.2
⅔	2	161.0	164.0	.463	6.38	83.5	28.9
Full	2	232.0	163.0	.453	6.24	86.5	29.5
Full	10	232.0	162.0	.445	6.13	86.5	30.2
Over	4	251.0	161.7	.441	6.08	87.0	30.5

Exhaust clear on all tests.

(*c*) Acceptance test, May 2–5, 1906. Triple-cylinder engine, 16×24″, 164 r.p.m. Kind of oil, distillate.

Load.	Duration, Hours.	B.H.P.	R.p.m.	Oil, Pounds per B.H.P. Hour.	Oil, Gallons per 100 B.H.P. Hours.	Economic Efficiency, Per Cent.
⅓	2	87.4	173.9	.558	7.5	23.9
⅔	2	156.7	168.9	.462	6.2	29.0
Full	2	224.0	164.8	.452	6.1	29.7
Full	10	224.0	167.2	.453	6.1	29.7
Over	4	254.0	163.6	.459	6.2	29.2

Engine drove air compressor.

(*d*) Acceptance test, Dec. 22–26, 1906. Triple engine, 16×24″, 164 r.p.m. Kind of oil, Whiting fuel oil; weight, 7.42 lbs. per gallon. Heating value, 19 500 B.T.U. per pound.

Load.	Duration, Hours.	B.H.P.	R.p.m.	Oil, Pounds per B.H.P. Hour.	Oil, Gallons per 100 B.H.P. Hours.	Economic Efficiency, Per Cent.
⅓	2	90.5	171.2	.556	7.5	23.5
⅔	2	162.9	166.8	.481	6.5	27.3
Full	2	230.3	165.4	.470	6.33	28.0
Full	8	230.9	165.9	.477	6.02	29.5
Over	4	258.5	165.5	.476	6.42	27.6

Allowances, experimentally determined, made for air compressor which was driven by motor:

⅓ load	13 H.P.
⅔ load	15
Full load	22
Overload	25

Exhaust was clear on all tests.

(*e*) Acceptance, July 30 and 31, 1907. Triple-cylinder engine, 16×24″, 164 r.p.m. Kind of oil, Whiting fuel oil, weight = 7.47 lbs. per gallon. Heating value, 19 500 B.T.U. per pound.

Load.	Duration, Hours.	B.H.P.	R.p.m.	Oil, Pounds per B.H.P. Hour.	Oil, Gallons per 100 B.H.P. Hours.	Economic Efficiency, Per Cent.
½	2	83.1	168.1	.558	7.47	23.3
⅔	2	157.8	166.8	.443	5.93	28.7
Full	2	228.4	164.5	.440	5.89	30.0
Full	8	227.6	164.1	.452	6.05	29.2
Over	4	254.9	162.3	.450	6.02	29.3

Air compressor driven by motor. Allowance made for this:

⅓ load	17 H.P.
⅔ load	19
Full load	20
Overload	21

The average of the above five tests on the 16×24″ three-cylinder engines seems to indicate that with crude oil we may expect the following results:

Table 98

Load.	Oil, Pounds per B.H.P. Hour.	Economic Efficiency, Per Cent. (Heat. Value taken at 19 000 B.T.U. per lb.)
⅓	.564	23.7
⅔	.464	28.8
Full	.447	30.1
12% over	.451	29.7

(*f*) Plant record, Traction Terminal Building, Indianapolis, Ind. Equipment: two 170 B.H.P. engines, three-cylinder, 14×21″, 200 r.p.m., two 120 K.W. D.C. generators.

Total K.W. hours for January and February, 1906	105 230
Total oil used for January and February	11 009.5 gallons
Average kilowatts per day	1783
Average oil per day	185.4 gallons
Average oil per 100 K.W. hours	10.4 gallons
Average oil per K.W. hour	.104 gallons
Average cost per K.W. hour, at 3 cents per gal. for fuel.	3⅛ mills
Rated K.W. capacity of plant	240 K.W.
Average kilowatts per hour	74.3 K.W.
Average load factor	30.54 %

Load: 3 electric elevators; 15 arcs; 3000 inc. lamps connected; 100 H.P. in motors.

(*g*) Plant record, Citizens Electric Light & Ice Co., Lebanon, Ind. Equipment:

One 225 B.H.P. 3-cyl. engine, 16×24″, 165 r.p.m.
One 120 B.H.P. 3-cyl. engine, 12×18″, 225 r.p.m.
One 150 K.W. A.C. generator } Operated in parallel.
One 85 K.W. A.C. generator }

Plant operates 24 hours per day, but day load is very small, not exceeding 5 K.W. Results for period from Feb. 11, 1906, to Feb. 28, inclusive.

Total K.W. hours	27 850
Total oil used	3 339 gallons
Oil per 100 K.W.	12 gallons
Cost of fuel per K.W.	$3\frac{6}{10}$ mills
Capacity of plant	235 K.W.
Average load	63.6 K.W.
Average load factor	27%
Load	
Number public arcs	95
Number private arcs	29
Number incandescent lamps connected	6 000
Motors on day circuit	8 H.P.

(*h*) Plant record, Wayne Paper Mills, Hartford City, Ind. Equipment:

Two 225 B.H.P. 3-cyl. engines, 16×24″, 165 r.p.m.
Two 150 K.W. A.C. generators operated in parallel.

Plant operates 144 hours per week.
Record for period covering 18 consecutive days.

Total K.W. hours	91 753
Total oil used	8 831 gallons
Average K.W. per day	5 609.6
Average oil per day	552 gallons
Average oil per 100 K.W.	9.84 gallons
Average oil per K.W.	.0984 gallons
Average cost per K.W. with oil at 3 cents per gal.	2.9 mills
Rated capacity plant	300 K.W.
Average K.W. per hour	233.33
Average load factor	77.8%
Cost of fuel per B.H.P. at engine, electrical efficiency 90%	2 mills

Load consists of induction motors of various sizes. Total installation about 500 B.H.P.

(*i*) Acceptance tests made on a 170 B.H.P. three-cylinder engine at Camden, N. Y., after one month of service showed the following results:[1]

Load.	Gallons of Oil per Hour		Pounds of Oil per Hour		Economic Efficiency, Per Cent.
	per K.W.	per B.H.P.	per K.W.	per B.H.P.	
$\frac{1}{4}$	.164	.110	1.21	.815	16.4
$\frac{1}{2}$	.112	.076	.83	.562	24.0
$\frac{3}{4}$	.101	.068	.747	.503	26.9
Full	.101	.068	.747	.503	26.9

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[1] Electrical World, June 2, 1906.

To determine the brake horse-power from kilowatts, the writer of the article assumed a uniform efficiency of 90%. This is probably too low at the rated load and too high at the lower load. The last three columns have been added by the present writer assuming the oil to weigh 7.4 lbs. per gallon and that its heating value is 19 000 B.T.U. per pound.

(*k*) Tests made on one 225 B.H.P. three-cylinder engine at the plant of the Kimberly & Clark paper mills, Kimberly, Wis. Engine drives one 150 K.W. A.C. generator. Tests were made by the Sargent Engineering Co. The figures in the following table give some idea of the fuel cost of compressing the air used for injection, together with some other data lacking in any of the other tests so far quoted. In this plant one unit of 120 B.H.P. was used to drive the air compressors for 900 B.H.P. of main engines. The table charges the 225 B.H.P. unit up with the proportionate part, that is one quarter, of the oil used in the compressor unit.

TABLE 99

	¼ Load.	½ Load.	¾ Load.	Full. Load.	Over-Load.
Date of run	1–5–06	1–5–06	1–5–06	1–13–06	1–13–06
Length of run in hours	1	2	2	4	.75
Revolutions per minute	179.1	177.1	174.5	169.1	161.2
Brake horse-power	68.15	133.7	194.9	249.7	263.8
Electric horse-power	54.5	113.6	170.6	224.8	237.5
Kilowatts	40.7	84.8	127.3	167.7	177.2
Pounds of fuel oil used in main engine per hour	42.75	57.1	82.5	106.37	126.66
Pounds of fuel oil used in compressor unit per hour	40.5	35.6	35.87	38.93	38.33
One-quarte. oil used in compressor unit per hour	10.12	8.9	8.97	.973	9.58
Total pounds of fuel oil used per hour	52.87	66.0	91.47	116.1	136.24
Pounds of fuel oil used per B.H.P. hour	.776	.493	.469	.464	.516
Pounds of fuel oil used per E.H.P. hour	.97	.580	.53	.516	.573
Pounds of fuel oil used per K.W. hour	1.29	.778	.718	.69	.77
Pounds of jacket water per hour	2033.5	2528.3	3324	3894.5	5306
Pounds of jacket water per B.H.P. hour	29.5	18.9	17.1	15.6	20.1
Average temperature inlet water	34.0	34.0	34.0	38.8	40.0
Average temperature outlet water	126.6	135.5	137.9	180.0	170.0
B.T.U. per B.H.P. hour	15103	9594	9127	9029	10041
B.T.U. per E.H.P. hour	18877	11287	10314	10041	11151
Thermal eff. of engine and dynamo $\frac{2545}{\text{B.T.U. per E.H.P.}}$	13.4	22.5	24.6	25.3	22.8
Thermal efficiency of engine $\frac{2545}{\text{B.T.U. per B.H.P.}}$	16.8	26.4	27.8	28.1	25.3
Percentage B.T.U. in water jacket	18.2	20.0	19.5	24.3	26.0
Percentage B.T.U. in exhaust, radiation and engine friction	65.0	53.4	52.7	47.6	48.7
Percentage B.T.U. in B.H.P.	16.8	26.4	27.8	28.1	25.3
Pounds of oil per 100 B.H.P. hours	77.6	49.3	46.9	46.4	51.6
Gallons oil per 100 B.H.P. hours	10.6	6.7	6.4	6.3	7.0
Gallons oil per 100 K.W. hours	17.6	10.6	9.8	9.4	10.5
Fuel cost of 100 B.H.P. hours in cents with oil at 3.64 cents per gallon	38.6	24.5	23.3	23.0	25.6
Fuel cost of 100 K.W. hours, cents	63.0	38.5	35.6	34.2	38.2

(*l*)[1] Tests made on the plant of the Prairie Pebble Phosphate Co., Mulberry, Fla., by Mr. R. E. Ludwig, between Feb. 7 and July 2, 1908.

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[1] See *Electrical World*, Oct. 3, 1908, for description of plant.

This plant consists of 7 units of 450 B.H.P., each direct-connected to a 300 K.W. Allis-Chalmers-Bullock alternating current generator. Each unit is composed of two 225 B.H.P. engines, standard dimensions, as shown on Plate XXXI. The generator is of the three-phase revolving field type operating at 60 cycles, 2300 volts, normal speed is 164 r.p.m. Each engine also drives a belted exciter.

The compressed air is furnished by five compressor units of the three-stage type, each unit being geared to a 75 H.P. Allis-Chalmers induction motor.

Each unit was tested separately, the load being obtained by water rheostat. The compressors were not driven by the unit under test but from the general bus-bars.

The oil used was desulphurized Texas fuel oil having the following characteristics.

Sp.gr. at 60° F.	.8689
Wt. per gallon at 60° F., lbs.	7.238
Flash point, ° F.	150
Water	None
Sulphur, %	.17
Acidity	None
Heating value, B.T.U. per lb.	19521

The oil consumption tests on Units 2, 4, 6 and 7, resulted as follows:

Unit No.	Oil Used per Hour.		Oil Used per 100 K.W. Hours.	
	Full Load (315 K.W.).	Half Load (160 K.W.).	Full Load (315 K.W.).	Half Load (160 K.W.).
2	196.6 lbs.	100.25 lbs.	8.62 gals.	8.66 gals.
4	187.38 "	102.8 "	8.22 "	8.88 "
6	195.5 "	102.0 "	8.57 "	8.81 "
7	187.0 "	102.5 "	8.20 "	8.85 "
Average	191.63 lbs.	101.89 lbs.	8.403 gals.	8.80 gals.

The efficiency of the generator was given equal to 93.3% at full load, and 90.10% at half load. The quantities for this plant were based on brake horse-power and cover the oil consumption of the compressor, but not that of the exciter. Hence the figures in the above table must be corrected.

The compressor power was found to be equal to 30 H.P. The exciter power consumption was determined at 8.7 K.W. or 13.7 H.P. The engine unit must be credited with the latter amount but should be charged up with the former. The net deduction is therefore $30-13.7=16.3$ H.P. $=12.2$ K.W. at the switchboard. The same allowance was made for both full and half loads. With the aid of the above generator efficiencies the results were next reduced to the contract basis with the following showing:

Load.	Net B.H.P.	Gallons of Oil per 100 B.H.P. Hours		Pounds of Oil per B.H.P. Hour from Tests.	Economic Efficiency, Per Cent.
		From Tests.	Guaranteed.		
315 K.W., Full	435.4	6.08	8.0	.440	29.7
160 K.W., Half	219.9	6.40	9.5	.465	28.0

The last two columns were added by the writer by aid of the data given above for the oil used.

Unit No. 4 was also tested at various other loads with the following results.

Load, K.W.	Oil Used per Hour, Lbs.	Oil per 100 K.W. Hours, Gallons.	Net B.H.P.	Oil per 100 B.H.P. Hours, Gallons.	Oil per B.H.P. Hour, Lbs.	Economic Efficiency, Per Cent.
0	56.00	.....	.....	....	....	....
80	77.33	13.36	107.4	9.94	.720	18.2
160	102.80	8.88	219.9	6.46	.468	27.8
240	147.33	8.48	330.8	6.16	.447	29.2
315	187.38	8.22	435.4	5.95	.432	30.2
330	202.00	8.46	456.6	6.11	.443	29.6
340	211.00	8.58	470.0	6.20	.449	29.0

Tests were also made to determine capacity and regulation. The average of all the capacity tests on the 7 units showed that the average maximum load carried was 473 B.H.P., which corresponds to an overload capacity of 5%. The regulation tests apparently consisted merely of determining the full and no-load speeds, the total difference between these two extremes was guaranteed not to exceed 10 r.p.m., or about 6%. The average total difference found on the tests was 8.5 r.p.m. or about 5.2%, all of the engines individually meeting the guarantee.

(*m*) Tests made on a 450 B.H.P. unit direct-connected to a 330 K.W. A.C. generator in the Silver Lake Power Plant at Pittsfield, Mass. The tests were made under the direction of Sibley College, Cornell University, by Messrs. A. Kennedy, J. L. Robbins, and M. duP. Lee, in April, 1907.

The auxiliary machinery consisted of a 15 K.W. D.C. exciter belted to an A.C. Stanley electric motor of 33 H.P.; two three-stage Norwalk air compressors belt driven by 25 H.P. Stanley electric motors; and a triplex Gould pump belted to a 5 H.P. A.C. motor, to supply jacket water.

The engines were of the same general dimensions as the 225 B.H.P. machines already described. For the purpose of the test, one half of the unit was disconnected and the other loaded by the generator, putting on service circuits as required. This of course made the generator operate at all times far below full load, but for the purpose of this test the generator was used merely as a brake, a calibration curve furnished by the Stanley Co. being used to reduce the K.W. output to the horsepower basis.

The crude oil used on the test was examined by Mr. B. S. Cushman of the Chemistry Department of the University, and showed the following characteristics:

Composition, C=86.97% by weight
H=12.50%
S= .30%
O= .23%
Heating value =19 272 B.T.U. per lb.
Sp. Gravity at 60° F.=.8776

The following table shows the results of the tests in detail, and but little explanation is needed. The indicators gave unreliable results when the engine was running at about ⅕ load, and this data is therefore not recorded. The reason for this apparently was that each cylinder obtained an ignition only now and then, the injection air driving the cylinder the rest of the time. As a result the indicator cards at this load showed scattered expansion lines, and it was not possible to get the I.H.P. accurately. For the remainder of the tests the cards were quite normal.

During normal operation the water used to cool the oil-injection valves was sprayed into the exhaust pipe for the purpose of keeping it cool. In order to determine the real exhaust gas temperature, this water supply was cut off several times for as long a period as appeared safe, hence the two lines recording exhaust gas temperature in the table.

An attempt was made to determine in two ways the ratio of air to oil used, from the analysis of the exhaust gas in conjunction with the analysis of the oil and by means of the volumetric efficiency of the cylinder as found from lower loop cards taken on each run. The results of these two methods of computation are also given in the table. Neither method is quite correct, since on the one hand uncertainty exists as to the temperature in the cylinder at the end of the suction stroke, while on the other no means was at hand to analyze the exhaust gases for unburned fuel. Attention is called, however, to the fact that at the high loads the agreement is close. The air theoretically required to burn each pound of the kind of oil used was 15.3 lbs., hence the air excess in operation was at least 55% at the highest load.

TABLE 100

	I	II	III	IV	V	VI
Duration, minutes	100	90	90	70	110	60
Barometer, inches Hg	29.28	29.28	29.28	29.28	29.28	29.28
Output generator, K.W.	26.61	58.53	85.39	116.26	146.37	162.51
Volts	2362	2424	2418	2443	2450	2438
Amperes	20.16	33.35	45.51	63.72	76.49	81.46
Power factor, %	55.7	69.0	77.8	74.5	78.2	81.8
Generator friction, wind., etc., K.W.	9.8	10.2	10.1	10.4	10.42	10.4
Armature loss, K.W.	.104	.29	.544	1.08	1.56	1.77
Total generator loss, K.W.	9.9	10.49	10.64	11.48	11.98	12.17
Efficiency generator, %	72.8	84.3	88.5	90.5	91.6	92.0
Engine output, gross, K.W.	36.51	69.02	96.03	127.74	158.35	174.68
Engine output, gross, H.P.	49.0	92.5	128.9	171.0	212.2	234.0
Imput. comp. motor, K.W.	16.0	15.58	15.54	15.6	15.8	15.54
Efficiency comp. motor, %	87.5	87.5	87.5	87.5	87.5	87.5
Output comp. motor, K.W.	14.0	13.62	13.6	13.64	13.8	13.6
Input pump motor, K.W.	2.79	2.72	2.8	2.9	2.8	2.7
Efficiency pump motor, %	80	80	80	80	80	80
Output pump motor, K.W.	2.23	2.18	2.24	2.32	2.24	2.16
Out. comp. and pump motor, K.W.	16.23	15.8	15.84	15.96	16.04	15.76
Engine output, net K.W.	20.27	53.22	80.19	111.78	142.31	158.92
Engine output, net H.P.	27.15	71.4	109.7	149.6	191.0	213.0
Jacket water, total cu.ft.	68.4	89.6	113.1	96.8	201.0	69.0
Jacket water, total, lbs.	4260	5600	7060	6040	12550	4300
Pounds water per hour	2560	3735	4710	5175	6850	4300
Temperature water entering, ° F.	50.3	51.53	52	52	52	53.47
Temperature water leaving	108	110.7	114.45	119.3	124.1	179.6
Water temperature range	57.7	59.17	62.45	67.3	72.1	126.13
Temperature room and supply air	66.5	67.9	68.15	68.15	68.2	69.3
Temperature exhaust gas with water	255.0	345.0	375.0	465.0	498.0	645.0
Exhaust gas range with water, ° F	188.5	277.1	306.85	396.85	429.8	575.7
Temperature exhaust gas, no water	257.0	360.0	412.0	525.0	642.0	705.0
Exhaust gas, range, no water, ° F.	190.5	292.1	343.85	456.85	573.8	635.7
Specific heat exhaust gas	.2410	.2418	.2432	.2442	.2457	.2479
Oil, total lbs	63.06	67.1	85.75	86.1	166.0	108.3
Oil per minute, lbs.	.6306	.7455	.95277	1.23	1.509	1.805
Oil per hour, lbs.	37.84	44.733	57.166	73.8	90.54	108.3
Revolutions per minute	167.57	165.8	166.0	167.0	167.7	167.0
Air tank pressure in atmospheres	65.9	67.2	68.0	71.8	73.5	71.85
Air tank pressure, lbs.	969	989	1000	1055	1080	1057

TABLE 100—*Continued*

	I			II			III			IV			V			IV		
Ratio $\frac{\text{air}}{\text{oil}}$ by weight, from exhaust gas analysis	84 26			55.67			50.30			41.44			26.44			24.00		
Same, from vol. eff.	69.92			55.76			42.86			34.61			28.88			23.93		
Indicator number	1	2	3	1	2	3	1	2	3	1	2	3	1	2	3	1	2	3
Maximum pressure, lbs. per sq. in.	640	494	568	620	540	572	630	521	573	640	536	601	640	539	606	704	590	640
M.E.P.	...	...	...	38.5	30.0	40.28	57.5	37.77	44.3	72.3	55.0	51.9	83.7	66.3	68.9	90.2	79.8	73
I.H.P. per cylinder	...	...	...	38.9	27.5	40.7	52.8	38.16	44.7	73.5	55.9	52.7	85.2	67.6	70.2	92.1	81.4	74
Total I.H.P.	....			107.09			141.1			182.15			223.0			248.30		
D.H.P. gross	49.0			92.5			128.9			171.0			212.0			234.0		
D.H.P. net	27.15			71.4			109.7			149.6			191.0			213.0		
Mechanical efficiency, gross	....			86.5			91.3			94.0			95.1			94.4		
Mechanical efficiency, net	....			66.2			77.6			82.5			85.6			86.0		
Friction H.P.	....			14.59			12.2			11.15			10.8			14.3		
Oil per I.H.P. per hour, gals.	....			.0571			.0555			.0555			.0557			.0597		
Oil per I.H.P. per hour, lbs.	....			.417			.405			.405			.406			.437		
Oil per gross D.H.P. per hour, gals.	.1055			.066			.0604			.0586			.0578			.0627		
Oil per net D.H.P. per hour, gals.	.19			.0853			.0723			.0669			.6045			.069		
Oil per gross D.H.P. per hour, lbs.*	.77			.482			.441			.427			.4225			.458		
Oil per net D.H.P. per hour, lbs.	1.385			.623			.53			.488			.470			.504		
Oil per 100 net D.H.P. per hour, gals.	19.0			8.53			7.23			6.69			6.45			6.9		
Fuel cost per 100 D.H.P. hour, cents.	68.5			30.7			26.0			24.05			23.2			24.8		
Heat balance:																		
Heat supplied per hour, B.T.U.	729000			864000			1100000			1420000			1745000			2080000		
Heat supplied per hour, %	100			100			100			100			100			100		
Absorbed in jacket, B.T.U.	147800			220000			294000			347000			293000			544000		
Absorbed in jacket, %	20.3			25.5			26.7			24.4			28.2			26.2		
Exhaust, B.T.U.	123100			182000			208000			291000			378000			421000		
Exhaust, %	16.9			21.1			18.9			20.5			21.7			20.2		
Thermal equivalent indicated work	....			273000			359000			463000			567000			632000		
Thermal equiv. indicated work, %	....			31.6			32.6			32.6			32.5			30.4		
Radiation and loss, B.T.U.	....			189000			239000			319000			306500			483000		
Radiation and loss, %	....			21.8			21.8			22.5			17.6			23.2		
B.T.U. per I.H.P. per hour	....			8060			7800			7800			7820			8380		
B.T.U. per gross D.H.P. per hour	14900			9310			8530			8240			8150			8820		
B.T.U. per net D.H.P. per hour	26700			12000			10180			9400			9100			9680		
Thermal efficiency, I.H.P.	....			31.6			32.6			32.6			32.5			30.4		
Thermal efficiency, gross D.H.P.	17.2			27.35			30.0			30.9			31.2			28.8		
Thermal efficiency, net D.H.P.	9.55			21.20			25.0			27.0			28.1			26.2		
Volumetric efficiency, %	95.5			92.6			89.0			92.6			94.4			93.6		

* Oil at 8.3 cents per gallon.

(*n*) Acceptance tests, March, 1908, Waco, Texas.

Three-cylinder 14×21″ engine rated at 170 B.H.P., r.p.m., 200. Engine had two wheels, 7 ft.×18×14″, each carrying a brake for the tests. Air compressor independently driven. Allowance made for this, 16 H.P. Kind of oil used, distillate, sp.gr. = 31° B.

Load.	Duration, Hours.	R.p.m.	B.H.P., Net.	Gallons of Oil per 100 B.H.P. Hours.	Pounds of Oil per B.H.P. Hour.	Economic Efficiency, Per Cent.
$\frac{1}{3}$	2	209.3	58.3	9.3	.68	.....
$\frac{2}{3}$	2	205.5	117.6	7.3	.53	.....
Full	2	202.5	172.3	6.4	.46	.....
Full	10	202.1	172.1	6.4	.47	.....
Over	4	199.3	190.3	6.6	.48	.....

(*o*) Acceptance tests, August, 1907, Coeymans, N. Y.

Three-cylinder 12×18″ engine rated at 120 B.H.P., 220 r.p.m., Engine had two 84×12″ wheels, one carrying brake for tests. Air compressor independently operated. Allowances made were:

For ⅓ and ⅔ load	16 H.P.
For full load	16.3 H.P.
For overload	16.5 H.P.

Kind of oil, distillate.

Load.	Duration, Hours.	R.p.m.	B.H.P., Net.	Gallons of Oil per 100 B.H.P. Hours.	Pounds of Oil per B.H.P. Hour.	Economic Efficiency, Per Cent.
⅓	2	228.8	39.3	9.6	.69	.....
⅔	2	226.6	82.0	6.9	.49	.....
Full	2	220.6	120.9	6.7	.48	.....
Full	10	222.2	121.9	6.6	.47	.....
Over	4	218.9	135.4	6.4	.46	.....

4. The Snow Steam Pump Works, Buffalo, N. Y. This company, the first in the United States to undertake the building of large gas engines, confines its product exclusively to the double-acting 4-cycle horizontal engine. Two types of these are built, Type B up to 500 B.H.P. in single tandem and up to 1000 B.H.P. in twin-tandem units, and type A up to 2500 B.H.P. in single-tandem and up to 5000 B.H.P. in twin-tandem units. Type B engine is also furnished as a single-cylinder or twin-cylinder engine. The following table shows the principal dimensions of the two types. Any of these sizes are built to operate on any of the following fuels: natural, producer, blast furnace, illuminating, coke oven, and all other industrial power gases.

TABLE 101

PRINCIPAL DIMENSIONS OF SNOW ENGINES

No.	Type.	B.H.P.		Diameter Cylinder.		Stroke.	R.p.m.
		S.T.	T.T.	Natural Gas.	Producer Gas.		
4	B	60	120	11″	12½″	12″	250
5	B	80	160	11½	12½	15	225
6	B	100	200	12½	14	15	225
7	B	125	250	13½	14¾	18	208
8	B	150	300	14	15½	21	197
9	B	200	400	15½	17	24	185
10	B	250	500	16	18	30	150
11	B	375	750	19½	22	36	130
12	B	500	1000	22	24	36	130
8	A	400	800	22	24	36	110
8½	A	500	1000	23	25½	42	105
9	A	650	1300	25½	28	42	105
10	A	800	1600	28	31	42	105
11	A	1000	2000	30	32	48	100
12	A	1250	2500	33	37	48	100
13	A	1500	3000	35	39	54	95
14	A	1750	3500	37	41	54	95
15	A	2000	4000	40	44	60	90
15½	A	2250	4500	42	47	60	90
16	A	2500	5000	43	48	60	90

There are two features in which the design of both types of engines differs radically from conventional European practice. The first is the adoption of the side crank in place of the center crank frame, and the second that the valves all open into a chamber at the side of the cylinder. All inlet valves and their gear are placed on top of these chambers and all exhaust valves and gearing on the bottom. The great advantage of this construction is that no part of the valve gear is below the floor, and the center of the cylinders can be kept low, making the engine rigid and steady. The foundation can be one continuous block instead of a series of isolated pieces. The foundation plate is continuous under all of the cylinders, and the latter slide upon this plate on machined surfaces, thus allowing of free expansion and contraction. The method of locating the valves in this manner makes the valve gear very simple, as may be seen from any of the following illustrations showing the gear. But one cam is used to operate both inlet and exhaust valves. The fear expressed that the location of the valve chamber at the side would effect the efficiency of combustion in an unfavorable sense, has apparently not materialized in practice. The further objection made that the cylinder as constructed would not be able to clear itself of any foreign material, such as dirt from the gas, incrusted oil, etc., also does not appear to have much weight.

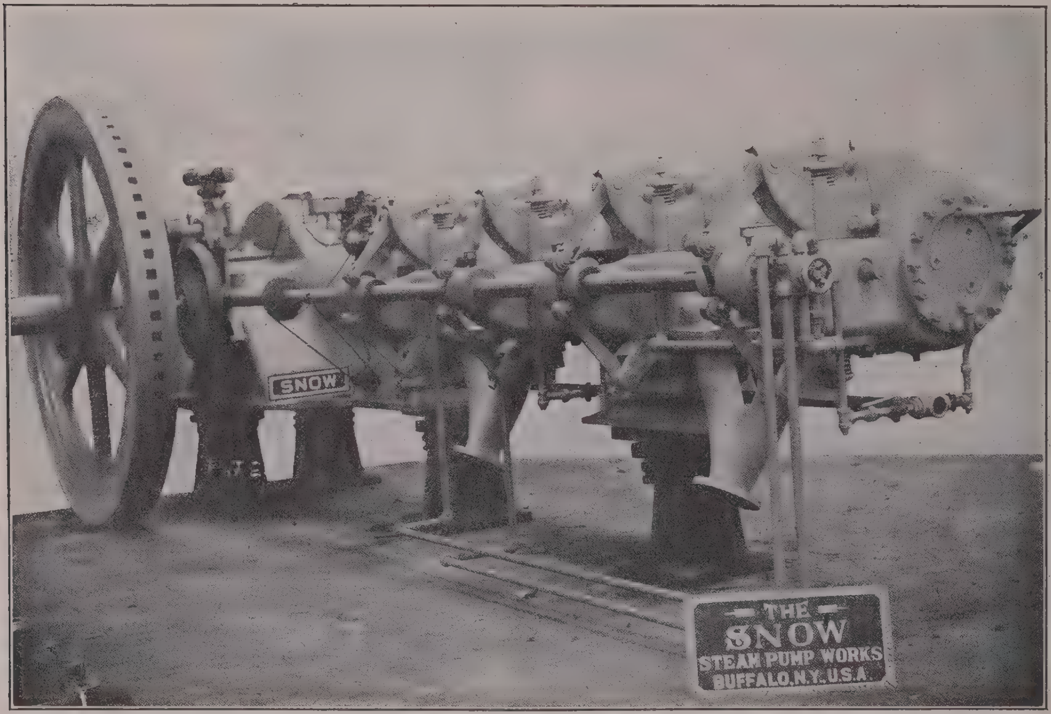

Fig. 605.

The constructive features of one of the small sized Type B engines are given in Figs. 605 and 606. For the larger sizes of this type the valve-gear construction is somewhat different, as may be seen from a study of Plates XXXIII, XXXIV, XXXV and Fig. 607. It will be noted that the speed regulation of this engine is effected by

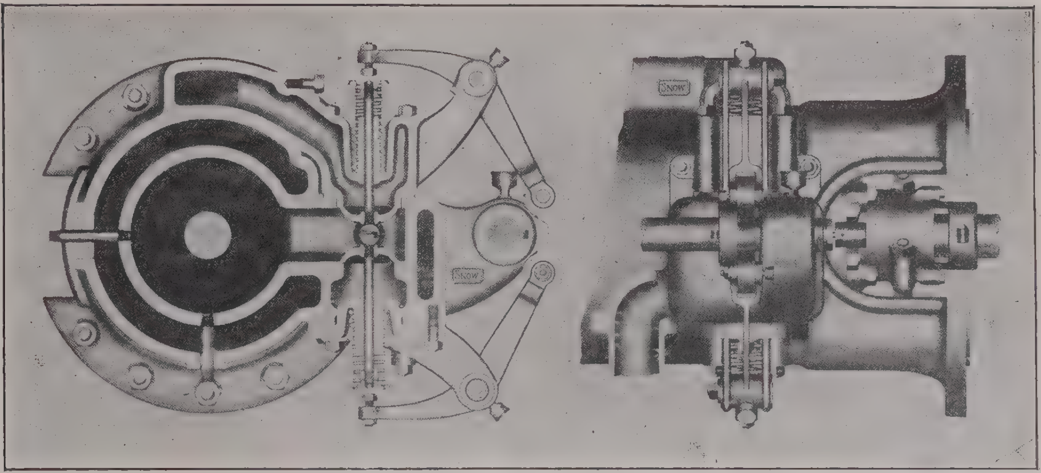

Fig. 606.—Valve Gear, Smaller Sizes of Type B Snow Engines.

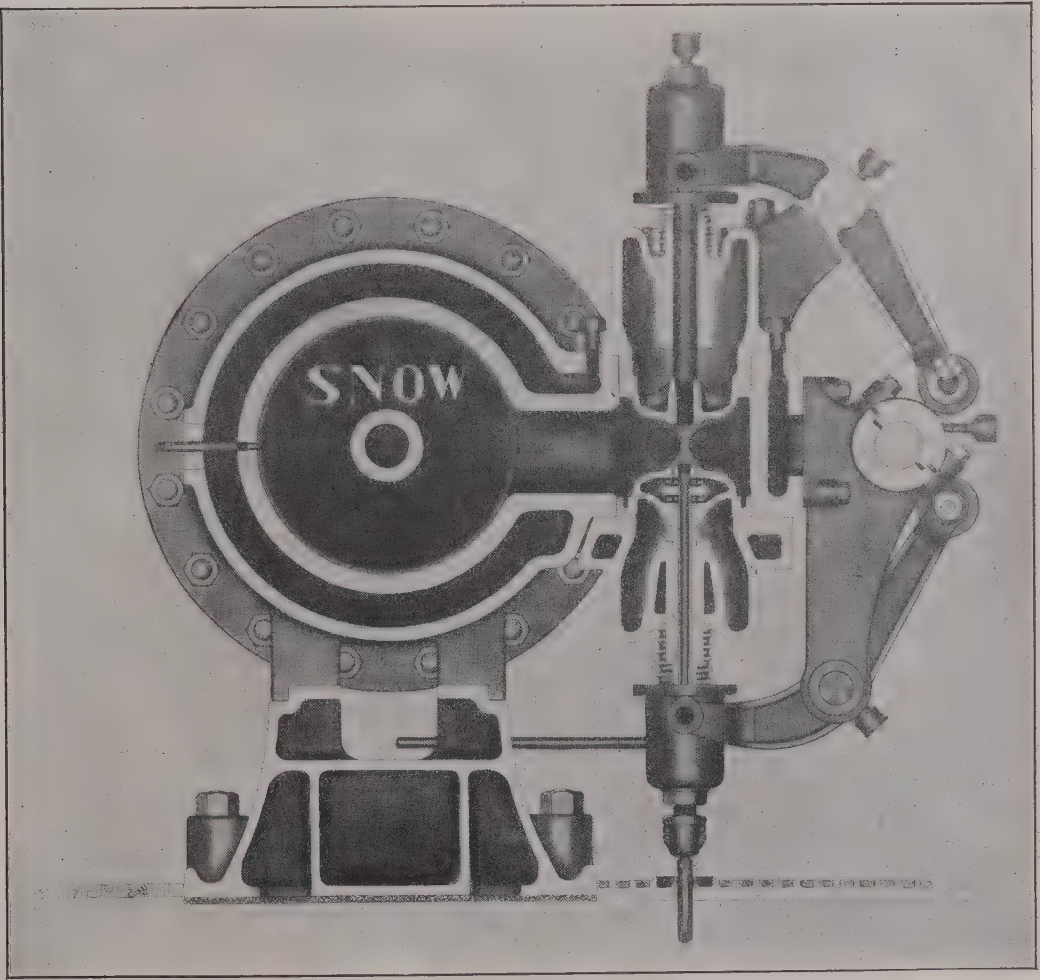

Fig. 607.—Valve Gear, Type B Snow Engines. 200 to 1000 H.P.

a centrifugal governor which controls a combined mixing and throttling valve. The latter is placed midway between the two inlet valves and serves both of them.

Type A engines differ from Type B in many points of frame and cylinder design, as will be seen by comparing Plate XXXVI with Plate XXXV. The general features of the valve gear on the two engines are the same except that the construction of the regulating and mixing valve in Type A differs with the service to which the engine is to be put. The first large installation of Snow engines consisted of four 5400 H.P. gas engines for the California Gas and Electric Corporation, San Francisco. One of these units which were used to operate Crocker-Wheeler generators, is shown in Fig. 608. In these engines the mixing and regulating valve was a sleeve valve concentric with the main inlet valve. Since then the use of this valve has been abandoned in favor of other constructions. Where the engine is to be used for a service which does not call for great refinement in speed regulation, such as pumping gas and water, or compressing air, the inlet valve gear of Type A is the same as that of Type B. Where close regulation is required, however as in electrical service, a separate drop cut-off mixing and regulating valve is placed just ahead of each inlet valve. The former construction of Type A gear (also Type B) is very well shown in Fig. 609, which shows three 1100 H.P. gas compressors, while the operation of the drop cut-off gear may be explained by the use of Figs. 610 and 611.[1] In the former figure, the air is admitted through *A* and the gas through *G* to the mixing chamber *M* above the main inlet valve *I*. Disks *A* and *G*, together with the short barrel *D*, form one casting. The disk *G* is provided with a taper seat and the length of the valve stem is adjusted in the block at the upper end so that both *A* and *G* will seat at the same time. The ratio of air to gas in the mixture is set once for all by adjusting the baffle disk *B* by means of the knurled nut *N*. The cut-off valve is operated as follows: Inlet valve *I* is opened by the rocker arm, Fig. 611, always at the beginning and end of the suction stroke. By means of the link *R*, Fig. 610, the main inlet valve stem operates a sliding block in a guide above the cut-off valve. To this sliding block is pivoted a latch, *L*, Fig. 611, which, when in position, engages with the block on the end of the cut-off valve stem and thus lifts this valve when the inlet valve *I* opens. To close the cut-off valve the governor controls the latch *L*, dragging it out of position by means of the drag link and the cam *C* shown in Fig. 611. The right hand end of the drag link curves around the governor shaft *S* and rests upon the journal box, thus holding the link in place as it slides back and forth. The cam *C* engages a lug on this link, thus displacing it and the latch *L*. At the moment the latter frees the block, the cut-off valve is closed by means of its spring, Fig. 610, and no more mixture is admitted to the cylinder. The cam shaft *S* rotates continuously at half speed. The point in the suction stroke at which cut-off occurs depends entirely upon the position of the cam, *C*, with relation to the crank-shaft, and this is controlled by the governor through a so-called "floating" bevel gear.

Operating Results. Figures relating to the performance of Snow engines in actual service are extremely scarce. The only results that the writer has ever seen were given in *Power*, July 14, 1908, and these do not give all of the data desirable.

Double-acting twin tandem engine at Ceres, N. Y. Cylinders 16″ diameter×30″ stroke, 150 r.p.m., driving 300 K.W. alternator. Fuel is natural gas estimated at 900 B.T.U. per cu.ft. The following table shows the gas and heat consumption for one

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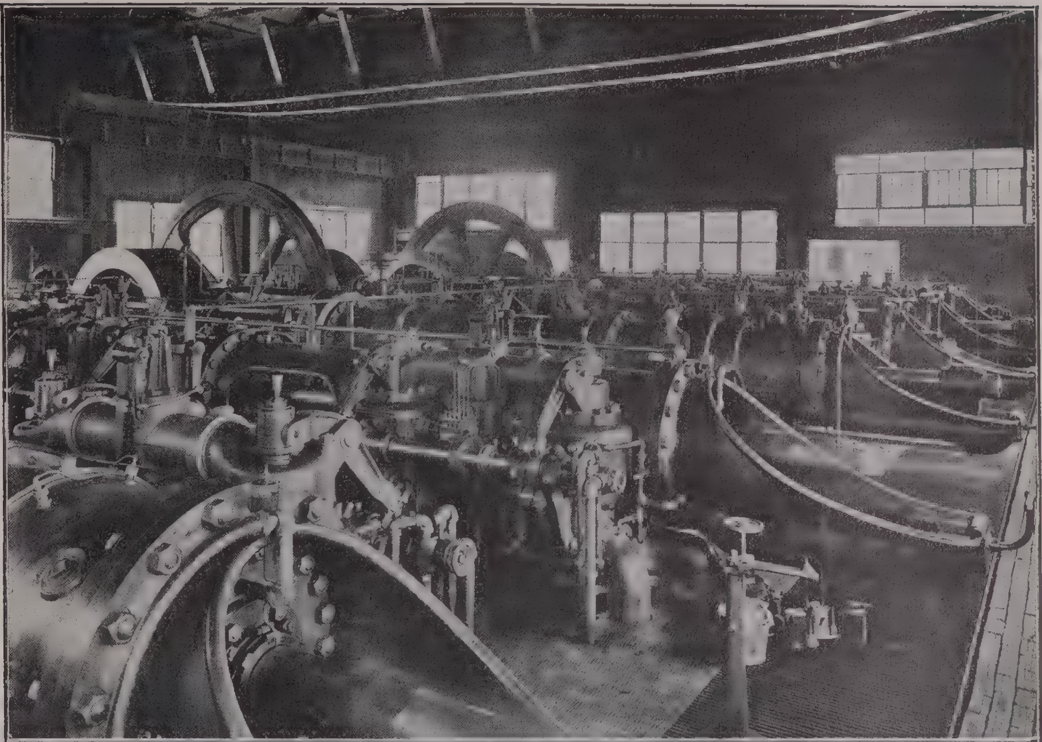
[1] *Power*, July 14, 1908.

Fig. 609.



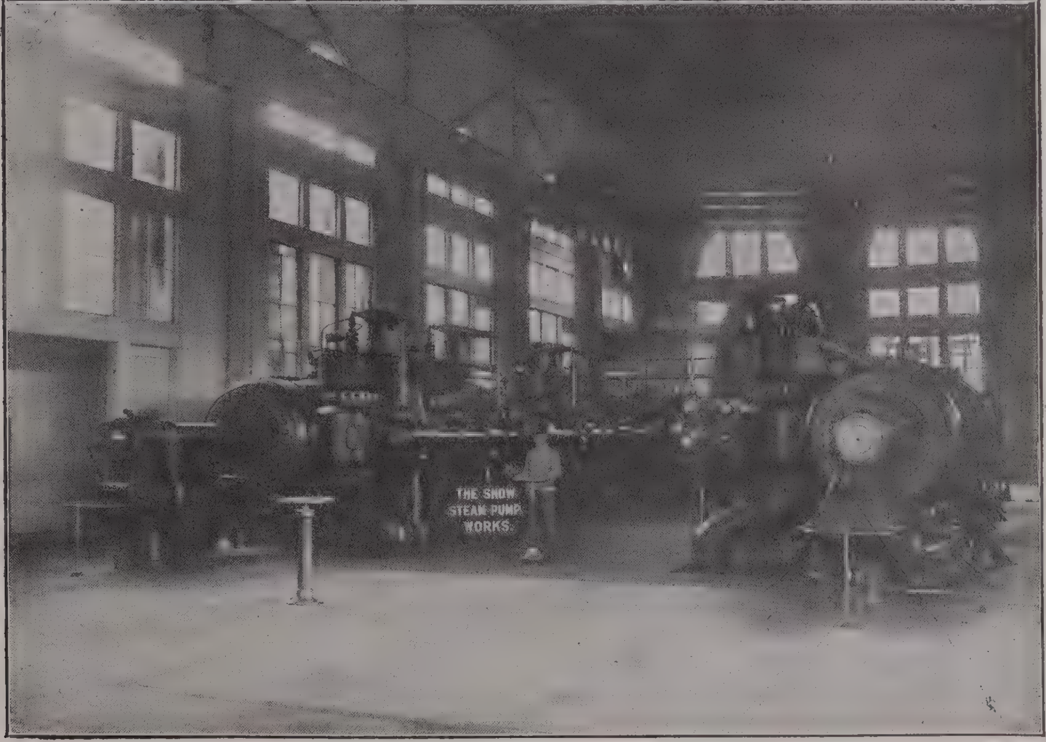


Fig. 608.

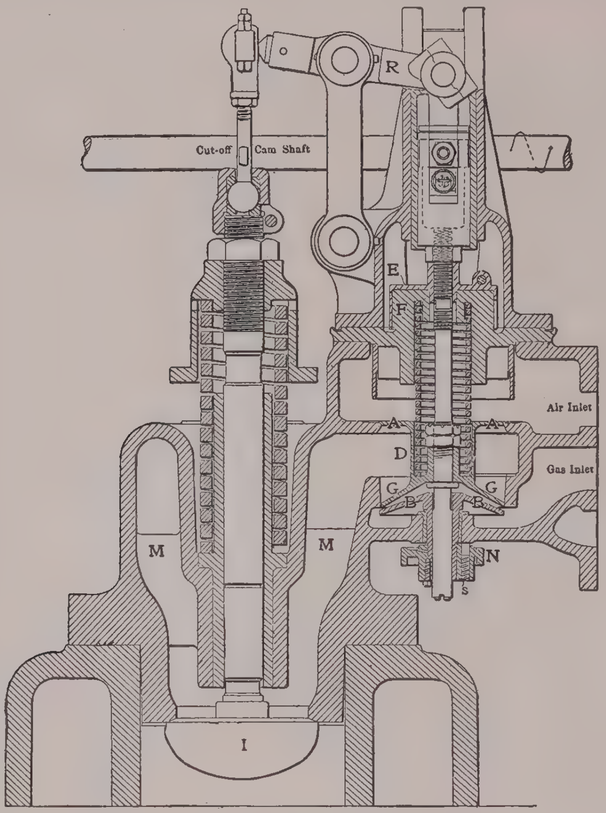


Fig. 610.

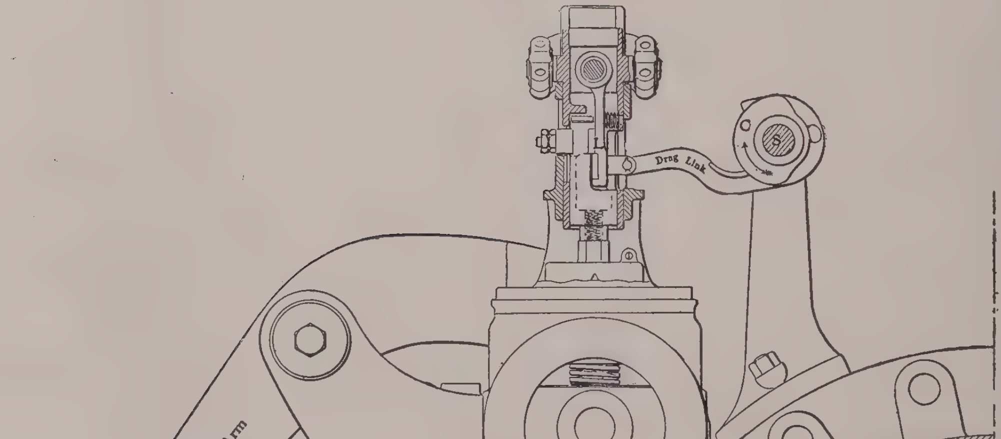


Fig. 611.

week. In view of the fact that the average load was only 240.8 K.W., the average heat consumption of 14 186 B.T.U. per B.H.P. hour, which corresponds to an economic efficiency of 17.9%, is not bad. To the table from *Power*, the writer has added the last two columns.

TABLE 102

Day.	Hours Run.	Kilowatts.	Average Load, Kilowatts.	Total Cubic Feet of Gas.	Gas per Kilowatt Hour, Cubic Feet.	B.T.U. per Brake H.P. Hour.*	Economic Efficiency, Per Cent.	Load, Per Cent of Full.
Sunday	21	4040	192.4	109960	27.2	16320	15.5	64.0
Monday	20	5050	202.5	127530	25.2	15120	16.8	67.5
Tuesday	21	4890	232.9	116400	23.8	14280	17.8	77.4
Wednesday	19	4730	248.9	106310	22.5	13500	18.9	83.0
Thursday	18	5050	280.5	111610	22.1	13260	19.2	93.2
Friday	20	5120	256.0	114060	22.5	13500	18.9	85.0
Saturday	19	5180	272.6	115060	22.2	13320	19.1	91.0

* Assuming 900 B.T.U per cubic foot and 1½ brake horse-power per kilowatt.

5. The Olds Gas Power Co., Lansing, Mich. *Type "K" Engines for Producer, Natural and Blast Furnace Gas.* In the production of an engine for the heavier gas power service, the Olds Company have adhered to the single-cylinder, single-acting, horizontal type of 4-cycle machine for all units up to 150 H.P. Also above this limit, a large number of plants have been installed, using a set of two or more of these machines, the reason for this policy being that in constructing engines of this type, absolutely no uncertain elements are introduced into the design. The various details of construction of this line have been repeatedly justified by their own engines in practical operation, and also by thousands of similar machines abroad.

As is usual with this type of construction, the cylinder jacket, or casting, is cast integral with the engine frame. From each side of the cylinder jacket casting two wing-like box girders extend to the main bearings. The outlines of this frame are well shown in the general view, Fig. 612 and the sectional view, Fig. 613.

Again, proceeding along well-tried lines, the cylinder proper is a separate casting, which is forced into the main casting from the head end. The advantages of thus being able to use a specially hard, close-grained iron for this particular casting, and also giving the liner perfect freedom to expand at the end, are now well understood in this country.

The cylinder head is still another casting containing the main valves, but for the sake of simplicity and strength contains as little of their various passages as practical.

The Olds Company anneal all of these heads to insure that initial shrinkage stresses are removed. As indicated by the above views, the construction is amply massive to withstand the heavy strains of the high-compression cycle.

A desirable feature which can always be attained in this construction is the absence of overhang of the cylinder. Thus, that part of the cylinder which takes the side thrust of the connecting rod is well within the bed.

In the larger sizes not only are the cylinder and head jacketed, but the exhaust valve and pipe as well.

There are a number of excellent design features embodied in these machines, among which may be mentioned the following.

The crank-shaft is an open-hearth steel forging, with the crank slotted from the solid. In the design of the shaft, it has been the policy of this firm to avoid, as far as possible, the overhung fly-wheel, and accordingly their standard machine has but one fly-wheel, which is designed of ample proportions to give the necessary cyclic regularity; and the shaft is supported by three bearings—two main bearings in the engine bed, and one bearing outside of the fly-wheel. The counter-balances for the cranks are both keyed and bolted to the cheeks. The shaft is of enough greater diameter at the fly-wheel to admit of key-seating without reducing the strength of the shaft. Two sets of tangential double keys are used to secure the fly-wheel to the shaft. The only feature of the connecting rod requiring special notice is the construction of the ends. The bearing metals are in detachable shells at both the crank and piston ends.

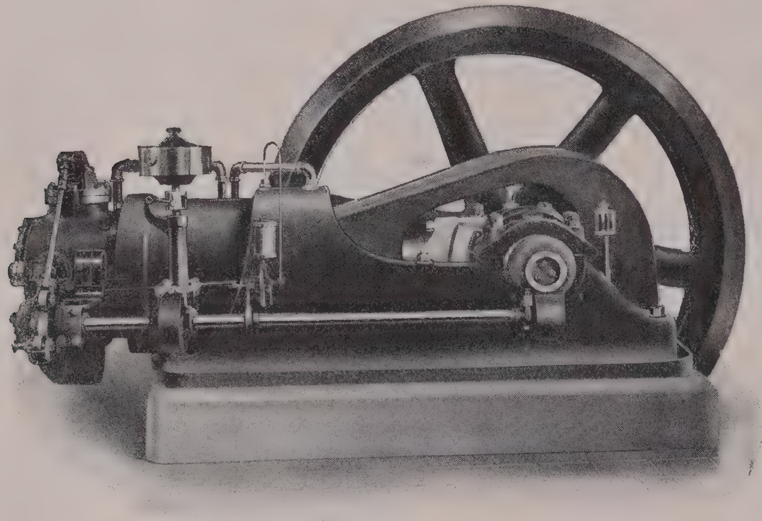


Fig. 612.—Olds Type "K" Engine.

As the trunk pistons of the large engines present an extensive rubbing surface, the internal friction from this source is very great on standard designs. The pistons of these particular engines, besides being equipped with seven regular piston rings, are provided with a number of dove-tailed grooves, filled with white, anti-friction metal. This material is thoroughly peened into place, and then machined to size. The piston itself is slightly smaller than the cylinder bore, so that the bearing comes on the alloy rings. The peculiar virtue of this metal is that it has all the anti-friction qualities of Babbitt, and at the same time is capable of withstanding the heat encountered in the gas-engine cylinder. In this way, the internal friction of the Olds engine is cut down to a minimum.

By reference to the general view, Fig. 612, it will be seen that the valve mechanism is driven from a lay shaft, which in turn is operated from the main shaft by means of Brown & Sharpe hardened helical gears. The governor has a positive drive from this same shaft, through similar gears. Both inlet and exhaust valves are operated by the same cam.

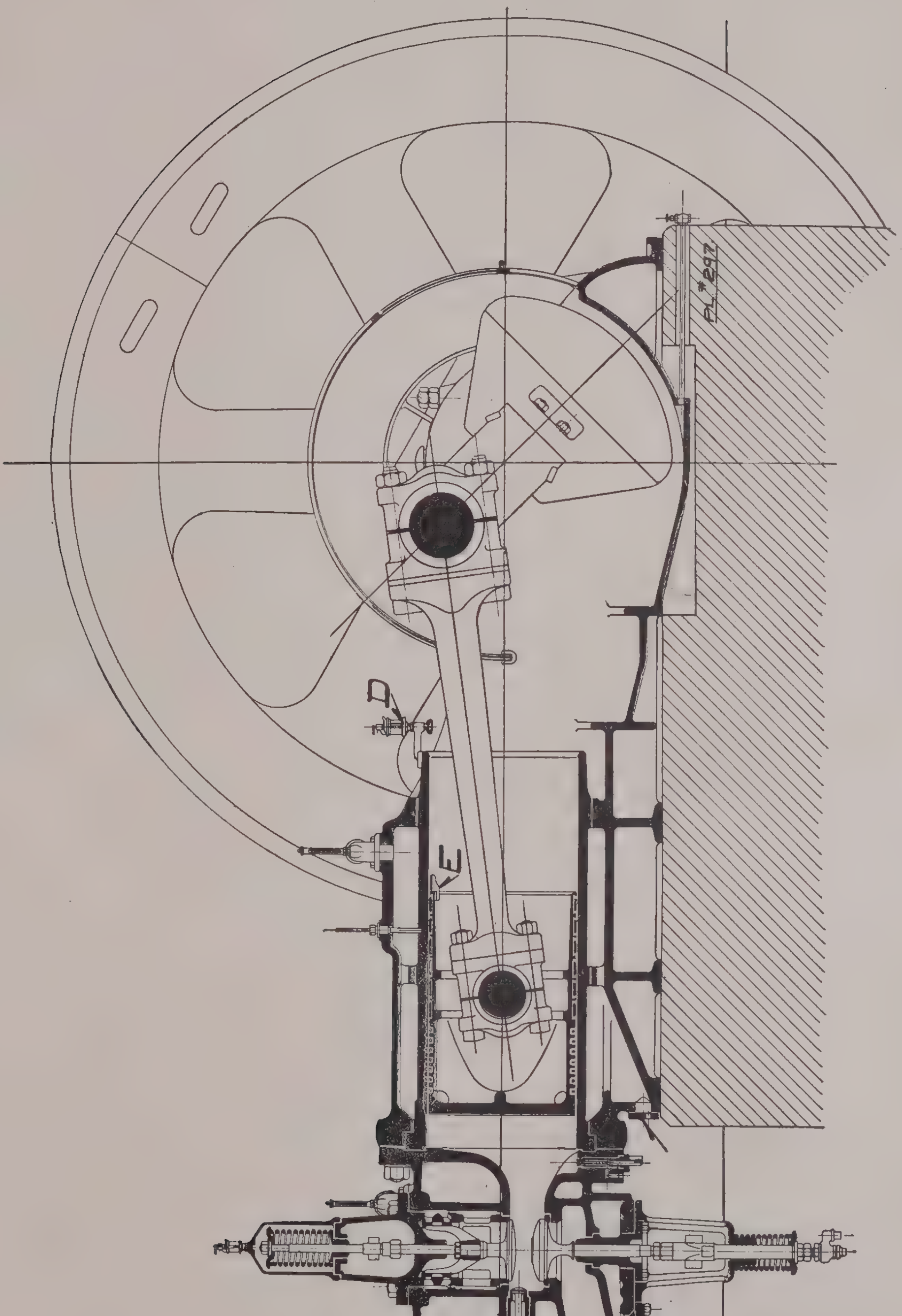


Fig. 613.—Olds Type "K" Engine.

By referring to Figs. 614 and 615, the valve mechanism between the cam and the two main valves will be readily understood. The exhaust valve is lifted through the rock-lever, *A*. The inlet valve is opened by means of the linkage *B* and *C*, and a rock-lever *E*. This lever has no stationary pivot, but rocks on the block *H*. The governor linkage is very simple. The centrifugal action of the governor weights tends to work the vertical lever *G* away from or towards the engine, according as the speed drops or increases; and this lever draws with it the fulcrum block *H*. Thus, when increasing load tends to lower the speed, the action of the governor draws the fulcrum block *H* away from the center of the engine, giving the inlet valve an increased lift. The gas is admitted through the passage *M*, while the air enters through *N*. The throttle valves in *N* and *M* admit of hand regulation of the mixture to suit the gas. The sliding sleeve *K* absolutely prevents the mingling of gas and air between strokes when the inlet valve is closed. The main valve springs are located outside of the bonnets and are instantly accessible for adjustment. Each valve stem is kept in alignment by two bushings well separated.

The starting valve is illustrated in Fig. 615. At *A* it is shown removed from the cylinder head; and at *B*, it is shown in position. The linkage for operating the valve can be readily traced. Referring to Fig. 614, the cam for operating the starting valve is shown at *Q* engaging a roller on the bell crank *R*. In Fig. 615 the letters *Q* and *R* are again used to designate these same parts. The method of connecting this bell crank to the valve is clear, without further lettering. In the view *A*, the starting handle is shown in position for operating the valve. The ordinary running position for this handle is shown by the dotted lines. When the handle is in this position, the weight of the horizontal arm of the lever *R* causes it to swing away from the cam, thus, entirely clearing it when the engine is running,

When it is desired to start the engine, the fly-wheel is barred around until the piston is just starting on the explosion stroke. In this position, the point of the starting cam, *Q*, is opposite the roller on the arm *R*. The gas and air hand valves being opened, the starting air is turned on at the stop valve, and the starting handle *S* is thrown to position shown in *A*, Fig. 615. This opens the air valve, admitting the compressed air to the cylinder, and starting the engine. The cam will admit air at every explosion stroke until the engine takes up its cycle. This usually occurs on the third revolution. After this, the starting air is shut off at the tank and at *S*. When the engine acquires sufficient speed, the exhaust cam roller *X* is pulled over to running position; thus giving the engine full compression, when it is ready for the load.

Ignition. The ignition system is of the make-and-break type, the current being furnished by a low tension Bosch magneto. The point of ignition can be easily adjusted during operation. Fig. 616 shows the method of operating the magneto and the position of the igniter block.

Lubrication. The main bearings on the standard engines are lubricated by ring oilers, the reservoirs of which are in the engine frame. All of these reservoirs are equipped with sight glasses to indicate the level of the oil. The crank pin is oiled from a collecting ring secured to the outside of the right hand crank cheek. The oil passes from this ring to the surface of the crank, through a hole drilled in the crank itself. Oil drops into the ring from the spout of a stationary sight-feed oiler. A sight-feed oiler, *D*, Fig. 613, supplies a wiper *E* from which the piston pin is lubricated. The piston and cylinder are lubricated by a sight-feed force pump, driven by an eccentric from the lay-shaft. The governor and main lay-shaft gears run in an oil

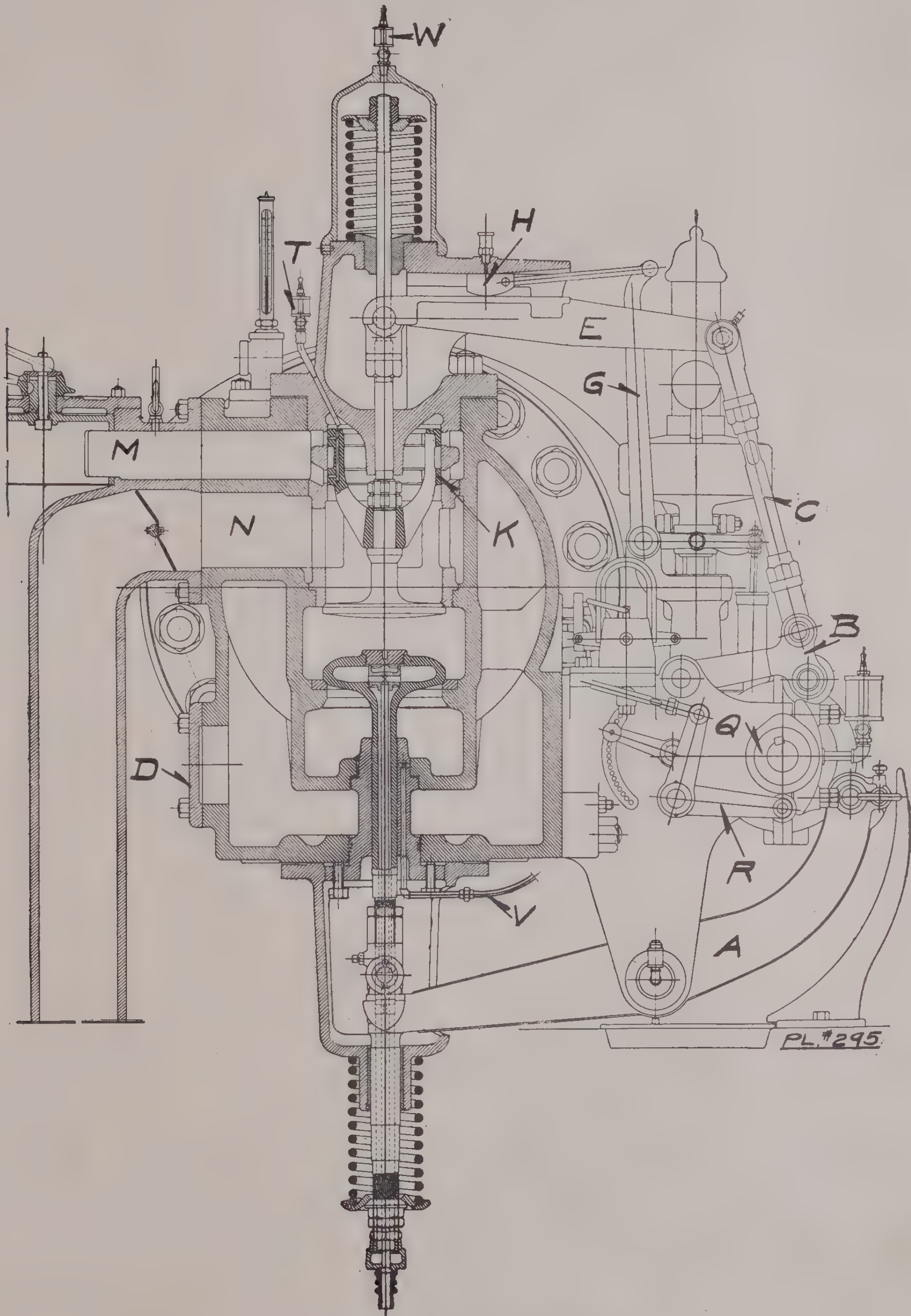


FIG. 614.—Olds Type "K" Engine.

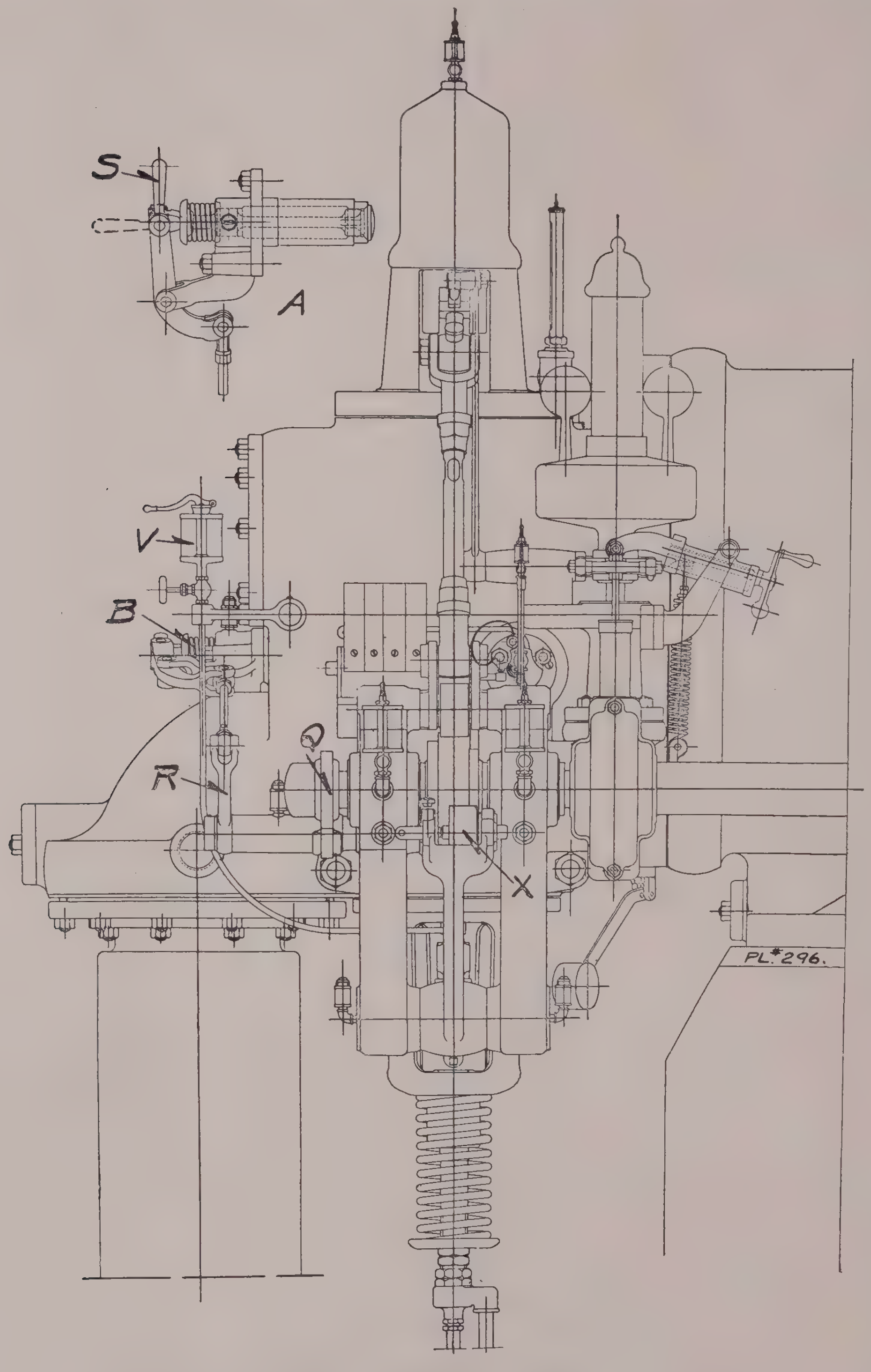


Fig. 615.—Olds Type "K" Engine.

bath. The hand-pump oiler, *V*, shown in Fig. 615 is used for lubricating the exhaust valve stem, through the oil pipe marked *V* in Fig. 614. In the same view the letter *T* indicates the oiler for lubricating the sleeve *K*. The oil supplied by the cup *W*, Fig. 614, works down the inlet stem, oiling the upper bushings, the rocker-arm yoke and the lower bushing in turn. Table 103 gives the principal dimensions of Olds Type "K" engines.

Olds Gasoline Engines. The Olds line of internal-combustion motors includes, besides the producer-gas machines just described, gasoline engines of from 8 to 50 H.P. They also manufacture a complete line of agricultural motors of from 1½ to 12 H.P. Both of these lines can be equipped with fittings for operating on city gas, or distillate. Finally, this company makes a line of small kerosene engines of from 3 to 12 H.P., which can be operated on gasoline, kerosene, or alcohol. So efficient is the mixer on this last type of motor that the company claims to be able to vaporize crude oil with it; but owing to the large amount of impurities in this fuel, does not recommend it.

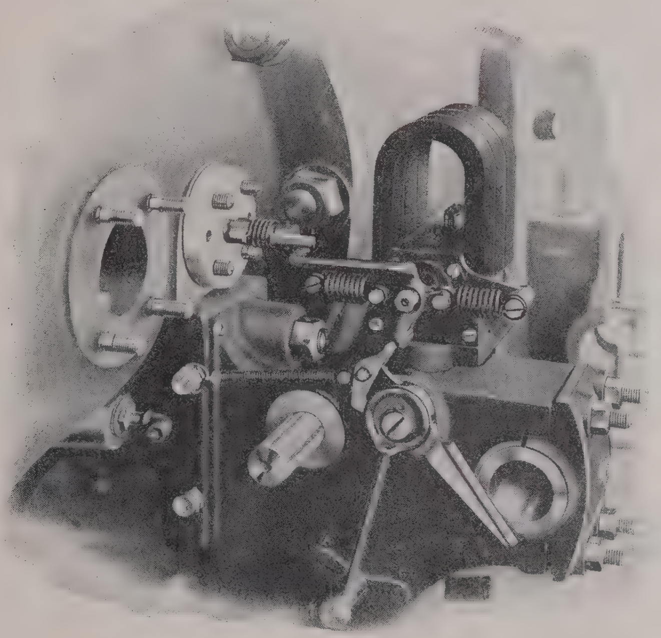


Fig. 616.—Ignition Gear, Olds Type "K" Engine.

The *Olds Type "A" engines* are built for a variety of purposes, for farm use, for direct connection to pumps, hoists, etc., and for general power purposes.

This motor is a horizontal, single-cylinder, four-stroke cycle machine, and for general utility is equipped with hit-and-miss regulation.

All engines of this type are furnished water cooled, either by means of the customary circulating tank, or by the new hopper jacket system, but, when desired, the 1½ and 3 H.P. sizes are also furnished air cooled for such special work as fruit spraying and the like.

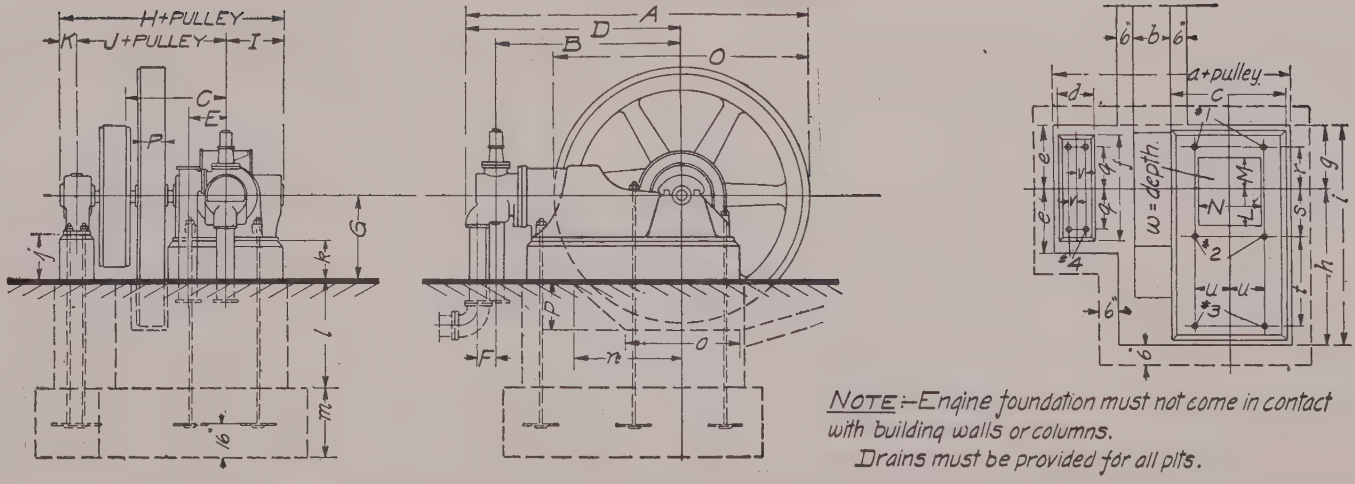
The jump-spark ignition system is used on the entire Type "A" line.

Instead of casting the frame in one piece, the Olds engines of this type have a

TABLE 103

OLDS GAS POWER COMPANY,

TYPE K ENGINES.



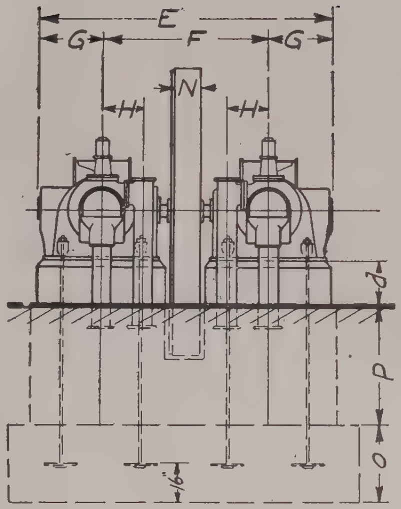
NOTE:—Engine foundation must not come in contact with building walls or columns.
Drains must be provided for all pits.

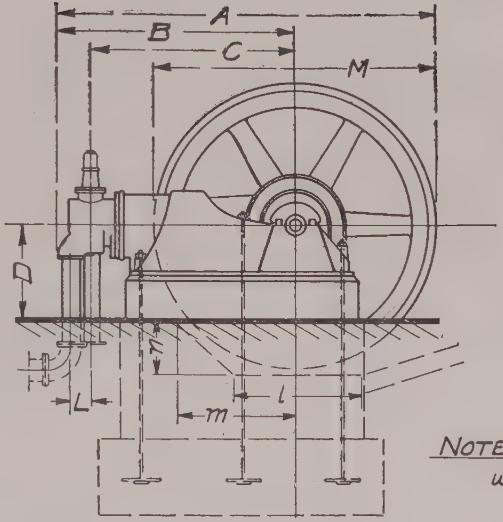
PL. #395.

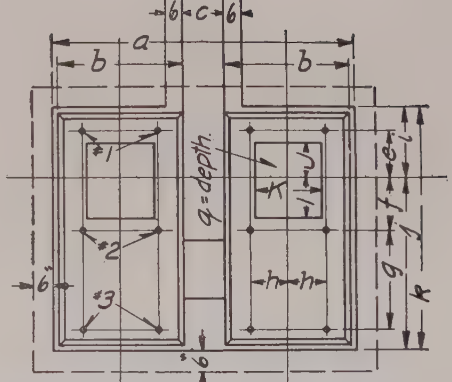
SIZE	H.P.	R.P.M.	GENERAL DIMENSIONS.														FLYWHEEL						PIPES					FOUNDATION DIMENSIONS.																							CUB. YD. CONCRT.	BOLTS.				
																	STAND.			ELECT.						WATER																											DIA.	LENGTH.			
			A	B	C	D	E	F	G	H	I	J	K	L	M	N	O	P	W	O	P	W	AIR	GAS	EXH.	INLET	DRAIN	a	b	c	d	e	f	g	h	i	j.	k	l	m	n	o	p	q	r	s	t	u	v	w			#1	#2	#3	#4
K-14	25	280	108	60	34	72	12½	6¾	27	65¾	19	41¾	5	0	0	0	72	9	4.5	72	9	5.1	3/4	3½	4	3/4	1	7½	16	37½	14	21	32¾	19	50½	73½	12¼	12¼	40	24	36	40	18	13½	13½	15¾	29¼	9⅜	0	0	8	1	66	78	66	66
	30	320	108	60	35	72	12½	6¾	27	65¾	19	41¾	5	0	0	0	72	10	3.6	72	10	4.5	3/4	3½	4	3/4	1	7½	16	37½	14	21	32¾	19	50½	73½	12¼	12¼	40	24	36	40	18	13½	13½	15¾	29¼	9⅜	0	0	8	1	66	78	66	66
K-18	40	240	142	80	45	94	15¾	8	30	84	25¼	54	6¾	0	0	0	96	10	7.8	96	10	9.8	1	4	5	1	1¼	90	20	50	18	27	44	25	67	96	13	10½	48	30	50	48	28	18	18	21	39	12½	0	0	16	1¼	80	96	80	80
	45	240	142	80	45	94	15¾	8	30	84	25¼	54	6¾	0	0	0	96	11	8.8	96	11½	11	1	4	5	1	1¼	90	20	50	18	27	44	25	67	96	13	10½	48	30	50	48	28	18	18	21	39	12½	0	0	16	1¼	80	96	80	80
	50	240	142	80	45	94	15¾	8	30	84	25¼	54	6¾	0	0	0	96	11½	9.8	96	11½	12	1	4	5	1	1¼	90	20	50	18	27	44	25	67	96	13	10½	48	30	50	48	28	18	18	21	39	12½	0	0	16	1¼	80	96	80	80
K-20	55	200	160	87	43	100	22¼	8	30	85¾	27	51½	7¼	18	12	26	108	11½	9.8	120	11½	12	1	4	5	1	1¼	94	21¾	44½	15	30	44	31	71	118	13	10	56	30	68	58	39	15½	23	20	45	15¼	6	7	20	1½	91	103	86	86
	65	200	160	87	43½	100	22¼	8	30	85¾	27	51½	7¼	18	12	26	120	11½	11	120	12	14	1	4	5	1	1¼	94	21¾	44½	15	30	44	31	71	118	13	10	56	30	68	58	39	15½	23	20	45	15¼	6	7	20	1½	91	103	86	86
	75	200	160	87	43½	100	22¼	8	30	85¾	27	51½	7¼	18	12	26	120	11½	13	120	12	16	1	4	5	1	1¼	94	21¾	44½	15	30	44	31	71	118	13	10	56	30	68	58	39	15½	23	20	45	15¼	6	7	20	1½	91	103	86	86
K-28	115	150	228	124	58	156	26	19	35	125¼	45¾	70½	9	24	20	30	144	13	28.5	144	14	36	1½	8	9	1½	2	136	27	63	0	45½	0	39	105	159	0	10	70	30	82	78	46	30½	27	33	60	20	6	6	50	2	115	127	105	102
	125	150	228	124	58	156	26	19	35	125¼	45¾	70½	9	24	20	30	144	13	30	144	14	37.5	1½	8	9	1½	2	136	27	63	0	45½	0	39	105	159	0	10	70	30	82	78	48	30½	27	33	60	20	6	6	50	2	115	127	105	102
	150	160	228	124	58	156	26	19	35	125¼	45¾	70½	9	24	20	30	144	14	36				1½	8	9	1½	2	136	27	63	0	45½	0	39	105	159	0	10	70	30	82	78	48	30½	27	33	60	20	6	6	50	2	115	127	105	102

NOTE:– W= weight in 1000 lbs. — All dimensions given in inches.

TABLE 103 (*continued*).







NOTE:– Engine foundation must not come in contact with building walls or columns.
Drains must be provided for all pits.

PL #398.

SIZE.	H.P.	R.P.M.	GENERAL DIMENSIONS.												FLYWHEEL						PIPES				
															STAND.			ELECT.						WATER	
			A	B	C	D	E	F	G	H	I	J	K	L	M	N	O	M	N	O	AIR	GAS	EXH.	INLET	DRAIN
K-14	50	280	108	72	60	27	106	68	19	12½	—	—	—	6¾	72	9	3.6	72	9	5.1	¾	3½	4	¾	1
	60	320	108	72	60	27	106	68	19	12½	—	—	—	6¾	72	10	2.7	72	10	4.5	¾	3½	4	¾	1
K-18	80	240	142	94	80	30	120	69½	25¼	15¾	—	—	—	8	96	10	6	96	10	9.8	1	4	5	1	1¼
	90	240	142	94	80	30	120	69½	25¼	15¾	—	—	—	8	96	11	6.6	96	11	11	1	4	5	1	1¼
	100	240	142	94	80	30	120	69½	25¼	15¾	—	—	—	8	96	11½	7.3	96	11½	12	1	4	5	1	1¼
K-20	110	200	160	100	87	30	120	66	27	22¼	18	12	26	8	108	11½	7.3	108	11½	12	1	4	5	1	1¼
	130	200	160	100	87	30	120	66	27	22¼	18	12	26	8	120	11½	8.5	120	11½	14	1	4	5	1	1¼
	150	200	160	100	87	30	120	66	27	22¼	18	12	26	8	120	11½	9.6	120	11½	16	1	4	5	1	1¼
K-28	230	150	228	156	124	35	138	96	46	26	24	20	30	19	144	18	20.7	144	18	36	1½	8	9	1½	2
	250	150	228	156	124	35	188	96	46	26	24	20	30	19	144	20	22.5	144	20	37.5	1½	8	9	1½	2
	300	160	228	156	124	35	188	96	46	26	24	20	30	19	144	20	27	144	20	37.5	1½	8	9	1½	2

SIZE.	H.P.	R.P.M.	FOUNDATION DIMENSIONS																	CUB. YD. CONCR.T.	BOLTS			
																						LENGTH		
			a	b	c	d	e	f	g	h	i	j	k	l	m	n	o	p	q		DIA.	#1	#2	#3
K-14	50	280	105½	37½	16	12¼	13½	15¾	29¼	9⅜	19	50½	73½	40	36	18	24	40	—	13	1	66	78	66
	60	320	105½	37½	16	12¼	13½	15¾	29¼	9⅜	19	50½	73½	40	36	18	24	40	—	13	1	66	78	66
K-18	80	240	119½	50	20	10½	18	21	39	12½	25	67	96	48	50	28	30	48	—	22	1¼	80	96	80
	90	240	119½	50	20	10½	18	21	39	12½	25	67	96	48	50	28	30	48	—	22	1¼	80	96	80
	100	240	119½	50	20	10½	18	21	39	12½	25	67	96	48	50	28	30	48	—	22	1¼	80	96	80
K-20	110	200	110½	44½	21¾	10	23	20	45	15¼	31	71	118	58	68	39	30	56	7	28	1½	91	103	86
	130	200	110½	44½	21¾	10	23	20	45	15¼	31	71	118	58	68	39	30	56	7	28	1½	91	103	86
	150	200	110½	44½	21¾	10	23	20	45	15¼	31	71	118	58	68	39	30	56	7	28	1½	91	103	86
K-28	230	150	159	63	33	10	27	33	60	20	39	105	159	78	82	48	30	70	6	60	2	115	127	105
	250	150	159	63	33	10	27	33	60	20	39	105	159	78	82	48	30	70	6	60	2	115	127	105
	300	160	159	63	33	10	27	33	60	20	39	105	159	78	82	48	30	70	6	60	2	115	127	105

NOTE:– O = weight in 1000 lbs.— All dimensions given in inches.

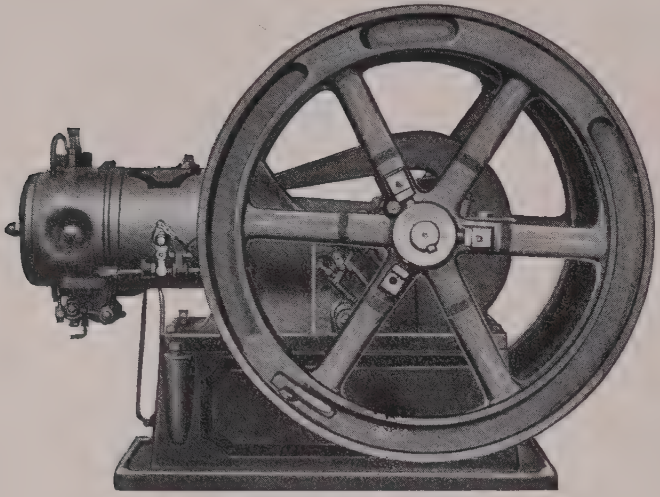


FIG. 617.—Olds Type "A" Engine.

cast iron sub-base, entirely separate from the engine bed proper, Fig. 617. This sub-base, being of box pattern, is utilized as a fuel reservoir. As it is made of cast iron of sufficient section to make a rigid base, it forms a substantial tank as well. In this way the hopper-jacket engine is completely self contained. When it is desired to mount such an engine on skids, there are but two parts to mount, the engine and the battery box; the fuel and cooling tanks are incorporated in the engine itself. The fuel is securely sealed from the air by an easily accessible screw cap, as in other makes.

The fuel mixer Fig. 618 is an application of the Venturi tube. This is a rather ingenious device, which not only atomizes the fuel thoroughly, but also raises it from the main tank to the auxiliary reservoir.

There are a large number of engines now on the market which dispense with the gasoline pump, but most of these do so merely by locating the mixer nozzle near the level of the main fuel tank. When the tank is full, the mixer tends to flood, and when the level is low, the mixture is too lean. The Olds mixer, on the contrary, raises the fuel to an auxiliary reservoir, from which it feeds at a constant head, no matter what the tank level may be.

The method of operation in this mixer will be apparent from an inspection of Fig. 618. *A* is the mixer tube, which delivers the mixture to the engine at *B*, drawing air from the outside through the opening *C*. The fuel is entrained from the nozzle *D* in the blast of inflowing air. Owing to the restricted area of the Venturi tube at *E*, the velocity of the air drawn into the cylinder by the engine piston is very high. This atomizes the gasoline issuing from the nozzle

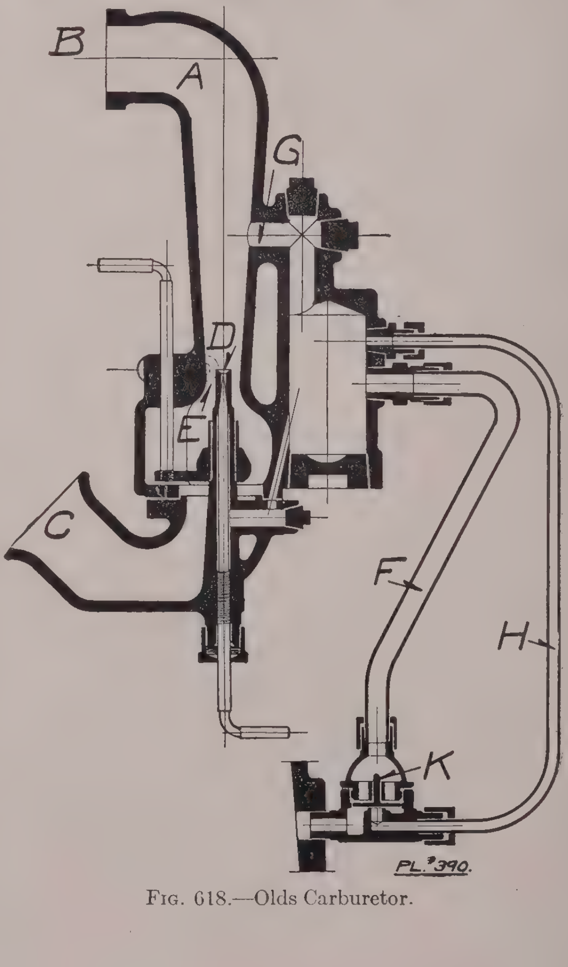


FIG. 618.—Olds Carburetor.

very thoroughly. The fuel always stands flush with the nozzle—any excess in the auxiliary reservoir will return to the main tank through the overflow pipe *F*.

On the suction stroke of the engine, the vacuum created in the mixing tube extends to the auxiliary reservoir, through the passage *G*. Both the pipes *F* and *H* being open, the atmospheric pressure in the main tank forces the liquid up the pipe *H*, filling the auxiliary reservoir. The reservoir cannot be flooded by the oil passing up the overflow pipe, as the pressure of the oil on the disk valve *K* raises the same, closing this passage during the suction stroke.

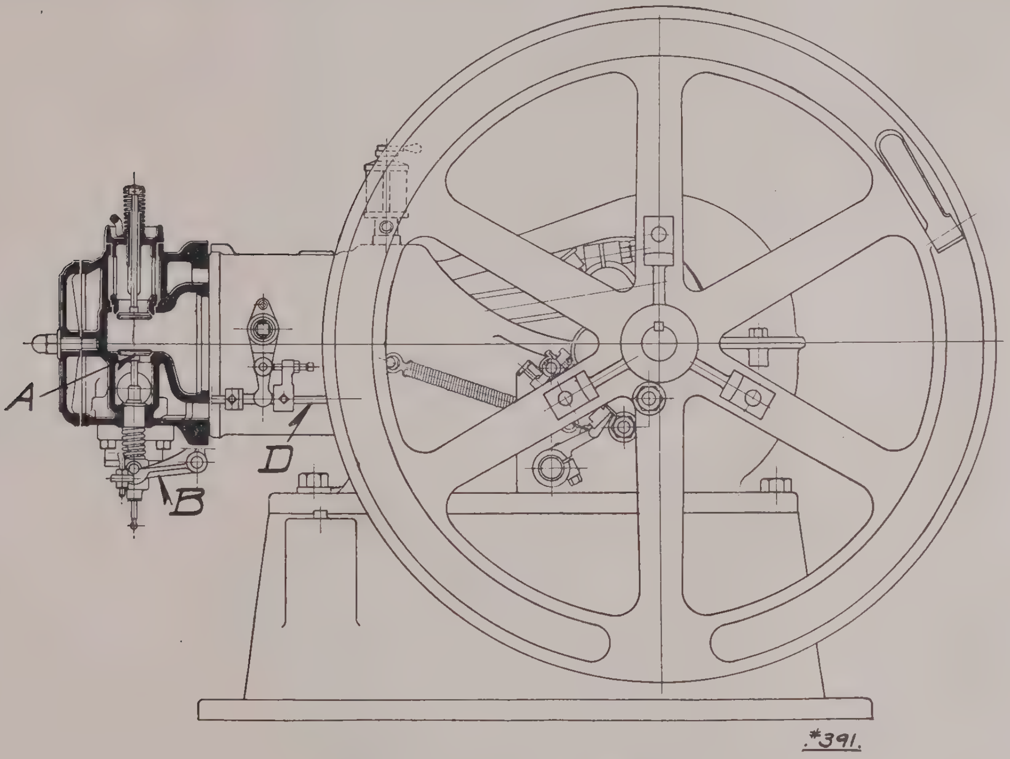


Fig. 619.—Olds Type "A" Engine.

At the end of the suction stroke, this valve will drop, however, allowing any excess in the auxiliary reservoir to return to the main tank. Thus, an absolutely uniform action is secured, no matter what the heights of the supply in the main tank.

The details of the construction in general and of the valve mechanism can best be understood by reference to Figs. 619, 620, and 621. From these it is evident that the piston, connecting rod, engine bed, etc. are designed in accordance with well tried standards.

The "two-to-one" gears are of the spur type, the teeth being machine cut, and the pinion being of bronze. The exhaust valve is operated from a cam which is secured directly to the driven gear, thus eliminating entirely the cam shaft and its torsional strain. The motion is transmitted to the exhaust valve *A*, Fig. 619, through a double bell crank *B* and a "pull-rod" *D*. The advantages claimed for this arrangement are that the gearing is enclosed in the bed casting, thus being protected from dust and grit; also, the larger gear, Fig. 622, touches an oil bath in the bottom of the bed, insuring lubrication. Furthermore, the link, or "pull-rod," between the valve and the cam is, by virtue of this arrangement, in tension instead of compression, and can therefore be made very light. Finally, this rod is provided with a simple adjustment, consisting merely of two nuts at the end, for taking up wear. This one adjustment takes up the wear in the entire mechanism, and is instantly accessible.

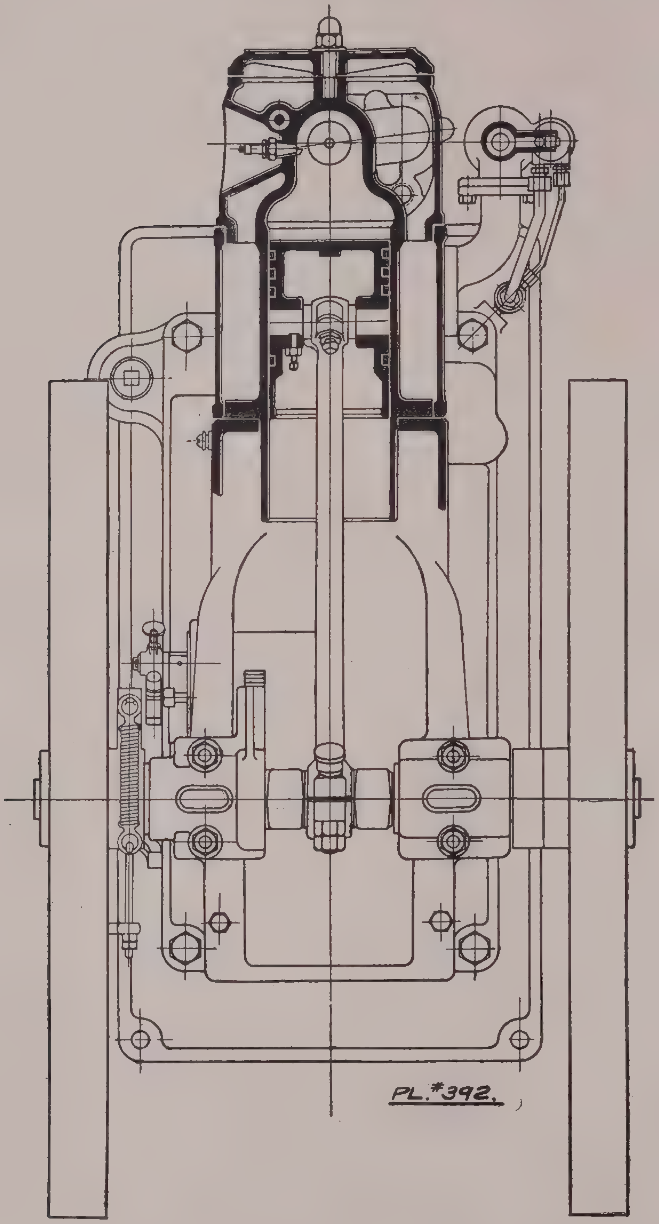


FIG. 620.—Olds Type "A" Engine.

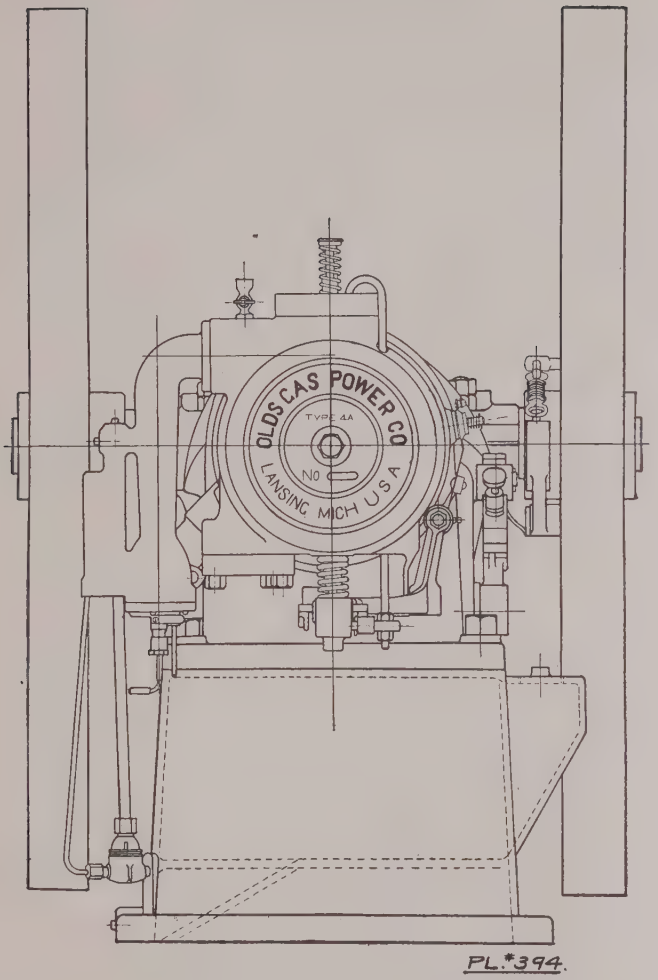


Fig. 621.—Olds Type "A" Engine.

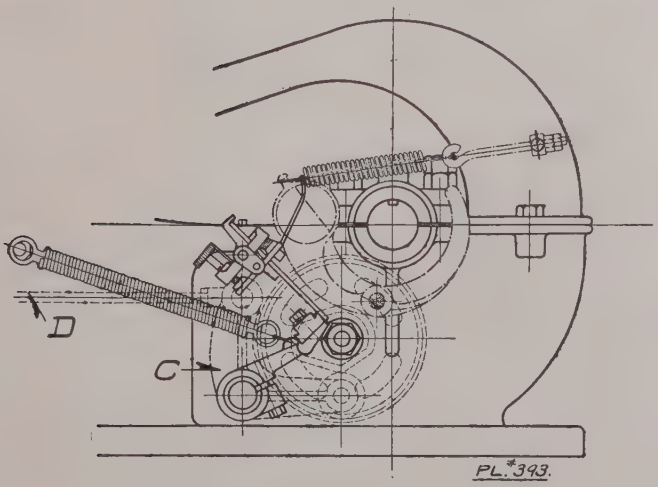


Fig. 622.—Governor, Olds Type "A" Engine.

The jackets of all these engines are made in a separate casting from the cylinder. Thus, the entire engine is not disabled in case of accidental freezing of the jacket water. The jacket alone is broken, and this can readily be replaced at a nominal sum. On all engines where the jacket is cast integral with the cylinder, such an accident entails the replacement of the entire cylinder, etc.

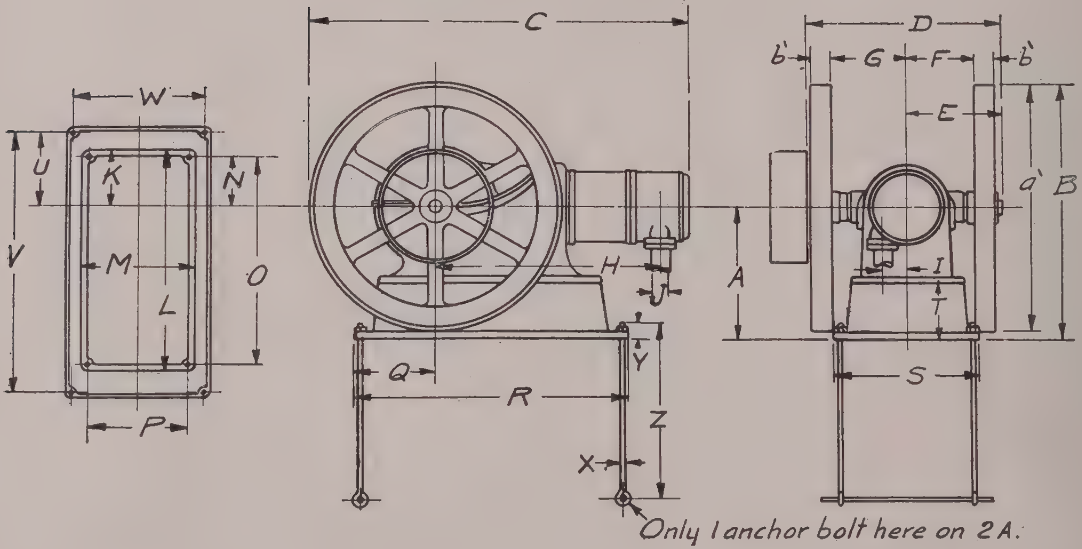
As stated before, Olds Type "A" engines are governed on the hit-and-miss principle. The inertia governor, located in the fly-wheel, is shown in detail in Fig. 622. It operates by interposing a pick blade, when the speed becomes too high, which prevents the bell crank *C*, shown in dotted line, from returning to its normal position. This blocks open the exhaust valve and causes a miss stroke. At the same time a rod passing from the exhaust-valve gear to the inlet valve, indicated in Figs. 619 and 621, puts an additional tension on the inlet-valve spring, and thus prevents the opening of that valve.

The following table shows the principal dimensions of Type "A" engines:

TABLE 104

OLDS GAS POWER COMPANY,

TYPE A ENGINES.



P.-#396.

TYPE.	H.P.	R.P.M.	ALL DIMENSIONS GIVEN IN INCHES.																												PULLEYS.			
			GENERAL DIMENSIONS.										ENGINE BASE.						SUB BASE.							ANCHOR BOLT.			WHEELS.		STAND.		SMALL'ST.	
			A	B	C	D	E	F	G	H	I	J	K	L	M	N	O	P	Q	R	S	T	U	V	W	X	Y	Z	a`	b`	DIA.	FCE.	DIA.	FCE.
2A	3	500	15 1/2	29 1/2	40 1/2	21 5/8	10 5/16	6 15/16	7 13/16	25 1/8	—	1 1/4	5 3/4	22	11	5	20 1/2	9 1/2	8 3/4	28	14	8 1/2	8	26 1/2	12 1/2	1/2	2	16	28	2	6	4		
3A	4 1/2	450	17 1/2	31 1/2	45 5/16	24 1/4	11 11/16	8 9/16	8 13/16	29 1/4	—	1 1/2	6 1/4	25 1/2	13	5 1/2	24	11 1/2	9 1/2	32	16 1/2	9	8 3/4	30 1/2	15	1/2	2	16	28	2 1/4	8	5	6	4
4A	6	425	19	35 1/2	49	26 5/16	13 1/2	9 1/8	9 9/16	27 1/8	3 3/16	2	7 1/4	27	14	6 3/8	25 1/4	12 1/4	10 3/4	34	18 1/2	10	9 7/8	32 1/4	16 3/4	5/8	2 1/2	20	33	2 5/8	10	5		
5A	8	400	20 1/2	38 1/2	53 9/16	26 11/16	13 7/8	9 9/16	9 11/16	30 3/16	3 3/16	2	6 7/8	28 3/4	14 1/4	6	27	12 1/2	10 3/4	36 1/2	18 3/4	11	9 7/8	34 3/4	17	5/8	2 1/2	20	36	3	12	6		
6A	12	380	21 1/2	41 1/2	63 1/4	33 1/4	17 1/4	10 11/16	11 1/2	35 5/8	3 3/4	2 1/2	9	35 1/2	18	8	33 1/2	16	13	43 1/2	23	9	12	41 1/2	21	3/4	2 1/2	28	40	3 1/4	18	6		

Table 105

SUMMARY OF RESULTS

	October 10–11.				October 11–12.				October 12–13.			
Duration of Test.	Hr.	Min.	Hr.	Min.	Hr.	Min.	Hr.	Min.	Hr.	Min.	Hr.	Min.
Engine running	8	9	7	53	10	38	8	1	8	2	10	2
Fires banked	15	56	16	..	13	18	16	..	16	..	14	2
Fuel.												
Coal used in running, lbs.	721.75		704.50		845.50		725.50		638.25		839.25	
Coal used in banking, lbs.	14.50		36.50		44.00		11.50		14.00		14.00	
Total coal-lbs.	736.25		741.00		889.50		737.00		672.25		853.25	
Salvage from screening, lbs.	239.00		266.50		368.00		311.50		185.00		202.00	
Salvage per cent of coal	39.50		26.00		41.50		42.10		28.20		23.60	
Coal per hour of running, lbs.	88.60		89.40		79.50		90.50		81.90		81.20	
Coal per hour of banking, lbs.	0.91		2.28		3.35		0.66		0.88		1.00	
Water.												
Water used in compressing air, both units, gals.	3 600.00				6 000.00				3 840.00			
Water used in blowing hot both units, gals.	8 200.00				11 600.00				9 800.00			
Time of blowing hot, minutes	17.50				24.50				20.50			
Water used in each vaporizer, gals. per hour	2.70				2.70				2.70			
Water in each vaporizer, per lb. of coal burned, lbs.	0.27				0.25				0.26			
Water used in each scrubber, gals. per hour	105.00				105.00				105.00			
Total water used in both scrubbers, gals.	1 900.00				1 680.00				1 960.00			
Water used in scrubber, per H.P. of producer, pounds per hour	7.30				7.30				7.30			
Water used in cooling each engine, gals. per min.	23.00				23.00				23.00			
Total water used in cooling engines, gals.	25 000.00				22 100.00				25 700.00			
Per cent of water pumped	1.80				1.80				1.90			
Total water used in plant, gals.	38 800.00				41 400.00				41 500.00			
Per cent of water pumped	2.90				3.40				3.10			

	October 10.		October 11.		October 12.	
Pumps.	No. 1.	No. 2.	No. 1.	No. 2.	No. 1.	No. 2.
Revolutions	15 652.00	15 417.00	19 440.00	15 768.00	15 668.00	19 676.00
Revolutions per minute	32.00	33.13	30.47	32.78	32.51	32.69
Discharge per revolution by displacement., gals.	40.62	40.62	.....	.....	.....	.....
Slip determined by preliminary test, %	6.20	1.70	.....	.....	.....	.....
Actual discharge per revolution, gals	38.10	39.93	.....	.....	.....	.....
Discharge per minute, gals.	1 219.20	1 322.88	1 160.91	1 308.91	1 238.63	1 305.31
Discharge per min., less water used in cooling, 24.80 gals	1 194.40	1 298.08	1 136.11	1 284.11	1 213.83	1 280.50
Total gallons pumped to reservoir	584 062.00	613 992.00	724 838.00	617 657.00	585 066.00	770 867.00
Pressures	No. 1.	No. 2.	No. 1.	No. 1.	No. 1.	No. 2.
Force main, ft.	164.80	168.20	166.00	167.00	166.70	166.70
Suction lift, ft.	12.69	11.90	11.65	11.19	13.42	12.67
Total	177.49	180.10	177.65	178.19	180.12	179.37

Duty. In million foot-pounds, based on plunger displacement, less slip and cooling water.	No. 1.	No. 2.	No. 1.	No. 2.	No. 1.	No. 2.	Average.
Per 100 lbs. coal fired during run	119.78	130.90	126.26	126.44	133.52	137.41	129.05
Per 100 lbs. of coal, including banking	117.43	124.46	120.01	124.47	130.74	135.15	125.38
Per 1 000 000 B.T.U. in coal fired during run	96.60	105.57	101.82	101.97	107.68	110.80	104.08
Per 1 000 000 B.T.U., including banking	94.70	100.71	96.78	100.71	105.44	109.00	101.20

Operating Results. The only figures available were obtained on tests of a pumping station at St. Stephen, N. B. This plant is equipped as follows:

Two Pintsch suction-gas producers, 135 H.P. each,
Two single-acting 4-cycle Type "K" engines, 125 H.P., 22″ dia., 28″ stroke, 150 r.p.m.,
Two-direct connected triplex pumps, capacity 1250 gal. per min., 13″ dia. × 12″ stroke, geared to run at 32 r.p.m.

The auxiliary apparatus consists of a compressor, driven by Pelton wheel and furnishing air for starting engines and producers.

The tests were made Oct. 10, 11, and 12, 1907. The following table contains the principal results. Indicator cards were apparently not taken.

The following are some additional figures contained in the report.

Coal used	Scotch Anthracite, Chestnut
Composition of coal, Moisture, %	1.17
Vol. matter, %	7.60
Fixed C., %	83.64
Ash, %	7.56
B.T.U. per lb. of coal	12 400
Coal used per water H.P., lbs.,	1.53
Coal used per B.H.P. (80% pump eff.) lbs.	1.22
Thermal efficiency on water lifted, %	13.4
Thermal efficiency on B.H.P., %	16.7
Temp. jacket water, inlet, °F.	44
Temp. jacket water, outlet, °F.	72
Jacket water per min., gals.	23.0
Heat loss in jacket water in % of heat in coal	37.0

It should be noted in connection with the thermal efficiency figures, that the plant was running at only about two-thirds capacity. The amount of jacket water is excessive, due to the low outlet temperature.

Olds Suction-Gas Producer. The Olds Gas Power Company owns the exclusive patent rights for the United States on the suction-gas producers of the Julius Pintsch Company, of Berlin, Germany. Fig. 623 is a sectional view of one of the 300 H.P. plants. The principal elements of the plant will be readily recognized; *A* being the generator, *B* the vaporizer, *C* the wet scrubber, and *J* the dry cleaners.

As in all suction types, the entire system operates under slightly less than atmospheric pressure; the engine drawing gas from the main, or exhausters, creating the draught through the apparatus. The cross-section area of the generator is especially large per rated horse-power, to insure low reaction temperatures, at rated loads, and consequent freedom from clinker troubles.

In addition, these producers are provided with a cone-shaped grate, on roller or ball bearings, which may not only be rotated, but also allows the fire to be reached by pokers through the bars. Four poke holes are provided just above the grate ring. These are of such size as to give ample freedom for the stoking bar, but still cut the heat loss by radiation to a minimum. When especially poor coal is used, additional facilities for barring are provided in the top of the producer. This top is not integral with the producer shell, but rotates on ball, or roller, bearings indicated at *F*. Thus, the barring hole *G*, and the coal hopper *D*, may be swung to any desired position.

The top plate and charging hopper are made gas tight by water seals. The gas leaves the producer by the pipe *E*, passes down through the vaporizer, and up through the scrubber. It reaches the two cleaners, *J*, through the pipe *H*. While these are contained in the same shell as the scrubber, yet they are entirely separated from it and each other by the partitions *MM*. The cleaners are furnished in duplicate to permit the

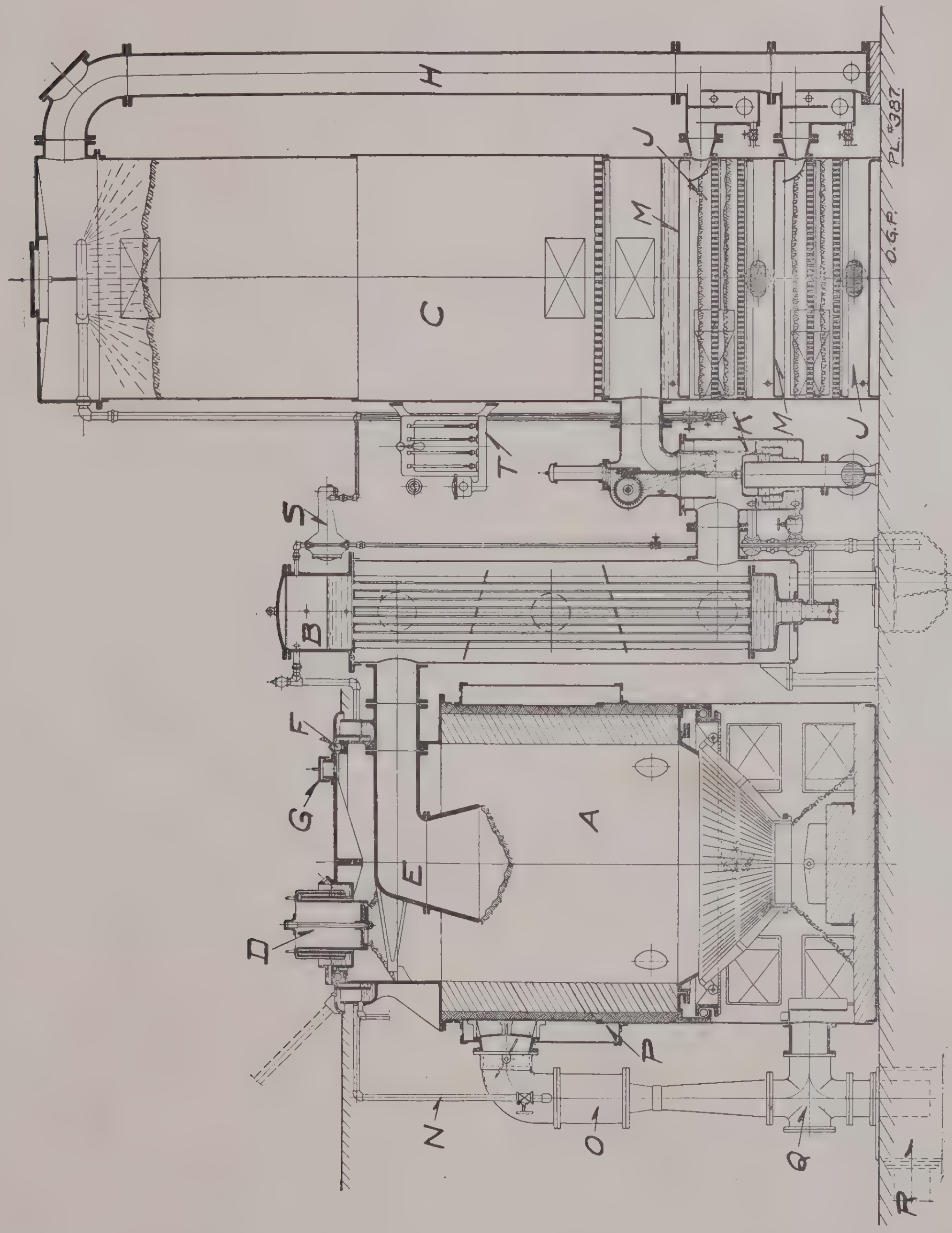


FIG. 623.—Olds Gas Producer Installation.

cleaning of one while the other remains in operation. The exit of the gas from the cleaners is shown at the back.

When it is desired to isolate the scrubber, etc., from the generator or producer, the bell *K* is raised by means of the rack and pinion. This action first closes the opening into the scrubber, and then only opens the vent-stack pipe. The entire nest of vaporizer tubes may be cleaned in place, or entirely removed from the cleaner, by loosening two flanges, and without breaking joints in connections between the producer, vaporizer and scrubber.

Perhaps the strongest feature about this producer is the method of securing a uniform mixture of steam and air for the fire. Steam from the vaporizer passes through the pipe *N* into the jet blower *O*; this blower, once set by hand, will deliver an absolutely uniform mixture of steam and air. The heated air is drawn from the producer jacket *P*, and is delivered with the steam, partly to the producer through *Q*, while any excess escapes through *R*. The jet blower supplies more air and steam than the gasification process requires, and as the producer draws its entire air supply from *Q*, it is obvious that whether the load be heavy or light, it will always get the same ratio of air and steam as long as the adjustment of the blower is unchanged.

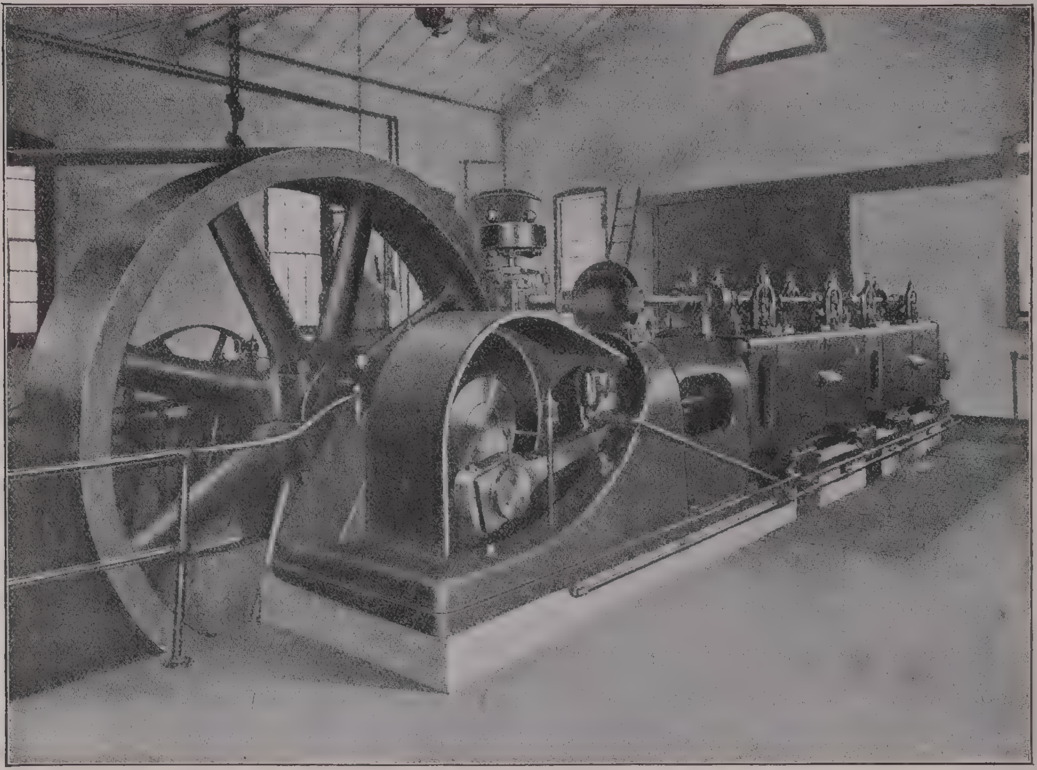


FIG. 624.—Front View of Riverside Gas Engine at the Watson-Stillman Plant, Aldine, N. J.

The water regulator *S* serves to maintain a constant level of water in the vaporizer, irrespective of steam consumption, and without waste. This regulator is connected with the vaporizer by a feed pipe delivering the water into the lower part of the vaporizer, and by another pipe connecting with the steam chamber of the vaporizer *B*.

The inlet and outlets of the producer, vaporizer, scrubber and cleaner are connected through piping with a series of water gauges on the gauge board *T*. Any irregularity in operation due to clogging, leaks, etc., in above apparatus can be immediately discovered by observing the vacuums shown by the gauges. Steam pressure in the vaporizer is indicated by a suitable gauge on the board.

Besides the engines so far discussed, the American market offers a number of other perhaps equally important machines, but owing to the comparative youth of the industry in this country, little information is available regarding their construction and next to none concerning their actual performance. It will therefore be possible only to give brief mention to any of these engines and to allow the reader himself to study the construction as far as that may be done from the pictures and cuts that were available. The information given was in some cases furnished in the shape of specifications by the engine company concerned, but in the majority of the cases is due to the columns of *Power*, and the Report of the Committee of the National Electric Light Association.

6. The Riverside Engine Co., Oil City, Pa.

Figs. 624, 625[1], and 626[1], together with the following specifications, will serve to show that the construction of this engine is unique in many respects. The company guarantees that the heat consumption of these engines on producer gas will not exceed 10 000 to 11 500 B.T.U. per B.H.P. hr., that the speed variation will not exceed 2% up to capacity, and that they will carry momentarily over loads of 10%.

SPECIFICATION OF RIVERSIDE HEAVY DUTY DOUBLE-ACTING TANDEM CLASS "F" GAS ENGINE

Cylinders and Sole Plate. The cylinders are cast in three pieces, two of which are alike, are of $\left\{\begin{array}{c}\text{cast iron,}\\ \text{semi-steel,}\\ \text{or steel}\end{array}\right\}$ according to size, and comprise the heads, valve chambers, rod housings, and combustion chambers. The third piece forms the cylinder barrel and jacket and is of air-furnace iron cast on end with a large sink head or riser. This piece, the most important part of a gas engine, is of absolute symmetry and extreme simplicity. Allowance is made for reboring the cylinder barrel, and this whole section can be as cheaply and more easily renewed than an independent cylinder liner. The three parts are fitted together with circular tongues and grooves, and held together and to the main frame by four steel tie bars, which take the tension strains.

The cylinders are mounted on a heavy cast-iron sole plate having a machined top surface, which keeps the cylinders in alignment and permits of their free expansion and contraction. By removing the distance pieces, the cylinder parts can be slid endwise on the sole plate, giving easy access to the interior of the cylinder, piston, and piston rings. The rings can be cleaned or changed, without disturbing the piston or rod.

The sole plate makes a perfect drip pan under all cylinders, keeping all oil drip from the foundation. The exhaust and inlet piping is attached to the sole plate, hence no piping except the water-jacket piping, has to be disturbed when the cylinders are moved. There is no overhead piping or wiring to interfere with a traveling crane

The sole plate contains a duct or passage for delivering fuel to each inlet valve and is tapped at exhaust passage for the connection of the exhaust piping.

Valves. Both inlet and exhaust valves are of the semi-balanced, water-cooled, poppet type, operated in a vertical position. All water to the cylinder jacket passes through the valves first, positively cooling them without any attention whatever.

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[1] *Power*, May 5, 1908.

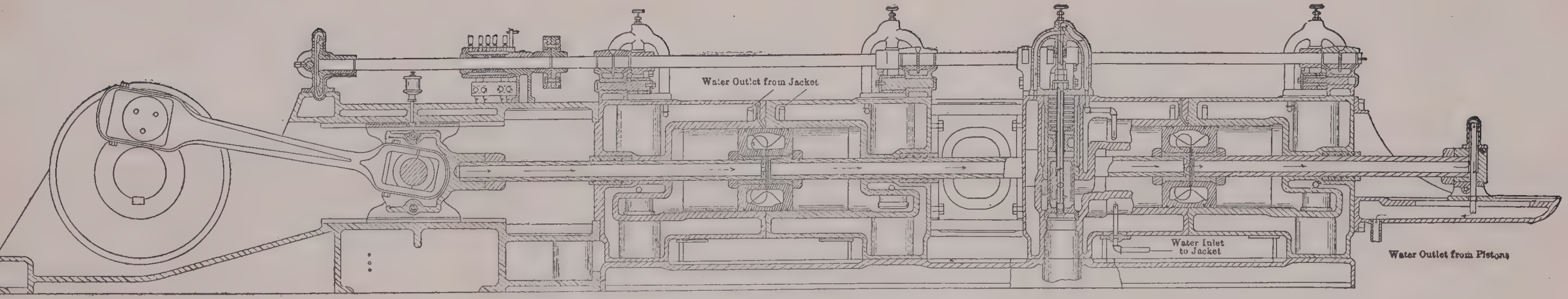
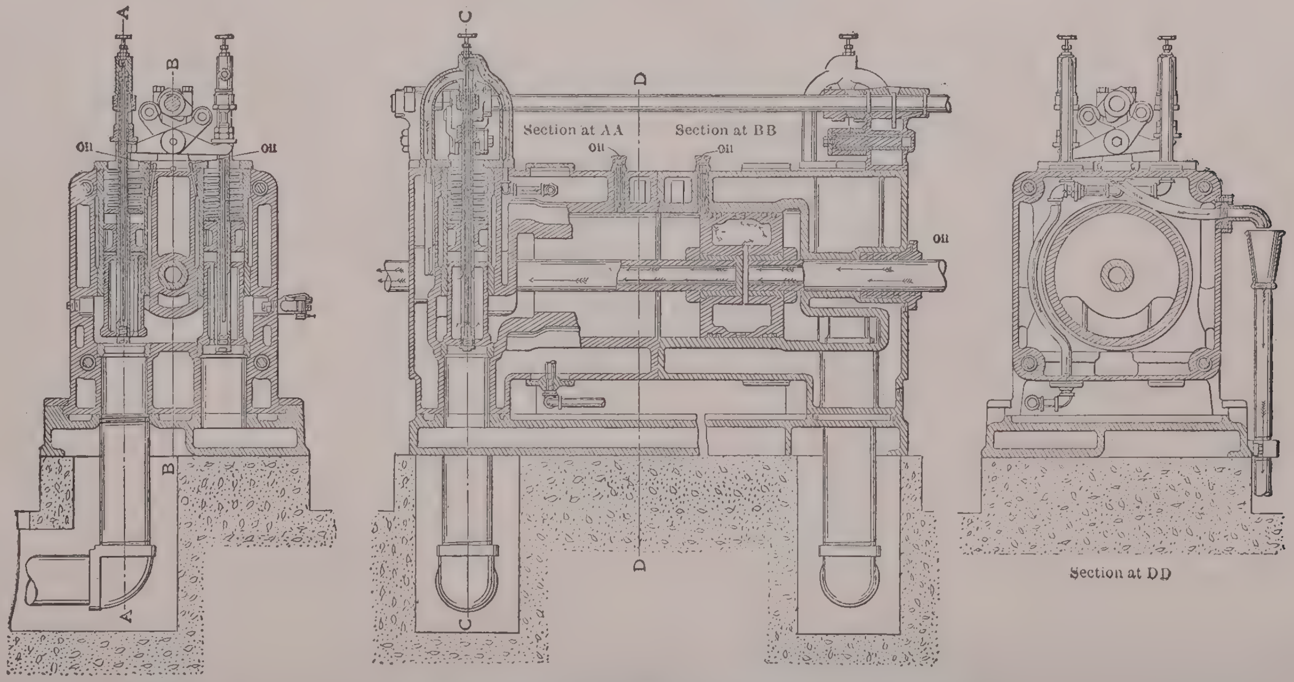


FIG. 625.—Longitudinal Section of Tandem Engine.



Section at CC

Section at DD

FIG. 626.—Section of Cylinder, showing Valve Arrangement.

The rise in temperature of the water in passing through the valves is only a few degrees, with the result that the valves are always cool, are not warped out of shape, are easily lubricated, never stick, and wear as well as the valves of a steam engine. The seats take a polish and run without regrinding for years.

The balancing pistons run in renewable liners and are lubricated by force feeds. These pistons provide a very large guide surface, assuring positive alignment for these valves, indefinitely. The valve seats are reversible, are easily renewed, and are located slightly below the bottom of the cylinder bore, so that any foreign substances are swept from the cylinder at each exhaust stroke. All valves are readily removed from the top of the cylinder without disturbing the cam shaft.

Piston and Piston Rod. Each piston and a section of rod form one solid piece. An enlarged end of the rod forms a shouldered steel center around which the piston is poured and to which it is welded. The rod end has rough grooves turned on it, making an intimate contact between the rod and piston. This construction also provides a steel center into which the adjacent section of piston rod is screwed. A locked-jam nut which is flush with the piston-face holds the sections securely together when they are in place.

The pistons and rod are supported at three points, viz., at the cross-head, the center rod support and at the end of the tail rod. These supports are fitted with adjustable babbitted shoes.

The construction of the *Riverside* piston rod is such that a piston and rod section in either cylinder can be removed, without disturbing the other cylinder or the connecting-rod, cross-head, or any part of the valve gear. Piston rods are made from steel forgings with water passages machined from the solid forging.

Piston and piston rods are water cooled, the water entering through a telescopic joint connected to the side of the cross-head. Circulation through each piston is positive and the overflow is so arranged that pistons are kept full of water. Water passages through the pistons and rod are large and easy, so that not over 10 lbs. pressure is required for circulation. The overflow is visible, hence the water cannot come to a boiling-point without attracting the engineer's attention.

Piston Rod Packing. Metallic packing is used in all piston-rod boxes. The packing rings consist of segmental cast-iron rings carefully turned all over and held by spring rings surrounding them. The ring sections are doweled in position and two completely assembled packing rings occupy the same groove in the gland. The gland is split in the direction of its axis and the halves held together by countersunk screws. The whole packing may be easily assembled around the rod and then pushed into the box and fastened.

Main Frame. The main frame or bed is of the heavy-duty, rolling mill type, with bored guides and main bearings cast in a single piece, and is of great weight and extreme rigidity. The main bearing jaws are carefully machined to receive the main bearings, which consist of Babbitt-lined shells according to size, $\left\{\begin{matrix}\text{semi-steel}\\ \text{or steel}\end{matrix}\right\}$. These are backed up for their entire length by adjusting wedges. The shells can be removed by raising the shaft slightly, and are adjustable for wear in both directions. The heavy main bearing cap locks across the top of main bearing jaws, and is provided with a large opening in its center for inspection.

The cylinder end of the main frame is squared and machined to fit the forward end of the cylinder, and has machined holes for receiving the tie bars which attach the cylinders.

The design of this frame, with its bored guideway, is such that a large part of the metal is above the center line, making it exceptionally stiff.

Cross-head. The cross-head is a heavy steel casting of the box type, fitted with adjustable top and bottom shoes, which are carefully turned to fit the bored guideway. The cross-head pin is carefully turned to size and fitted in the cross-head with a taper at both ends. It is drawn up and held by bolts passing through a holding plate which covers the end of the pin.

Two keyways are machined in the pin at 90° from each other. A single key holds the pin fast, relative to the cross-head, and the two keyways allow the pin to be rotated one quarter turn and so distribute the wear evenly over the surface.

The shoes are hung on pins, within eccentric bushings, and cannot be set out so as not to bear uniformly for their entire length. The crank-pin boxes are heavy steel shells, lined with genuine Babbitt. The cross-head boxes are of heavy phosphor bronze.

Crank-shaft. The shaft is of the side crank built-up type, and is machined all over. The bearings are ground and are of generous size, with liberal fillets for change of diameter, and there are no square cornered shoulders. The crank-disk is a heavy counterbalanced steel casting, with the crank pin cast integral, pressed onto the main shaft by hydraulic pressure and doubly keyed. The crank-shaft is guaranteed against breakage during the life of the engine.

Fly-wheel. The fly-wheel is of proper weight, with the face and edges of the rim turned to run true, and barring-over holes are provided.

The wheel is made in complete halves, and held together at the rim by heavy steel "T" links shrunk in the side and by bolts through lugs on the inner side of the rim. The wheel halves are held together at the hub by heavy steel bolts and the wheel is keyed rigidly to the shaft.

Valve Gear. The valve gear of the *Riverside* engine is the simplest ever applied to a multiple-cylinder engine, and consists of a single shaft mounted on top of the engine, running in self-oiling bearings. This shaft runs at one-half the speed of the main shaft, and carries the inlet and exhaust cams, cams for operating the oil pumps, and the timer for the ignition system. Power is transmitted to the inlet and exhaust valves by an inlet and exhaust lever hung on a single pin. All cams are keyed rigidly to the cam shaft. This construction makes a minimum number of joints subject to wear, although ample adjustments are provided for taking up any wear.

Governing. The speed of the engine will be maintained uniform within 2% either way from the mean under normal operating conditions, but might momentarily exceed this should full load be thrown on or off instantaneously. In no case will the speed exceed a safe limit. The speed is controlled by a sensitive carefully balanced enclosed type self-oiling centrifugal governor operating a double-beat balanced disk valve, which throttles the mixture, thus varying the mean effective pressure to suit the load and giving a uniform number of impulses regardless of the load. The mixture of air and gas can be varied at will and is controlled from one point so that all cylinders get an absolutely uniform mixture. This method of governing results in as uniform speed as in any system, but is without the complexity and necessitates none of the necessary accurate adjustments at each inlet valve which are incidental to a cut-off-gear.

Lubrication. The cylinders, valves, and stuffing-boxes are supplied with oil from force-feed lubricators. Lubrication to the cylinders is timed so that the oil is fed between the snap rings on the piston; thus the oil is swept over the entire surface of the cylinders and not fed into the cylinder at such a time as to burn up and do no work. Sight-feed oilers are provided for all other bearings except the cam shaft which is lubricated with ring oilers.

Ignition. Ignition is by an improved system, consisting of magnetically operated spark plugs in each cylinder. The sparking points are in series with the magnet coils which produce the break. This gives a positive external indication by the movement of the armature as to whether the spark takes place within the cylinder. The inductive resistance of the coils augments the strength of the spark which is extremely long and hot.

The timing is done electrically. No gearing or mechanical trips are used. Well insulated stranded wires, running through iron-armored conduits, lead from the timer to each spark plug.

The timer is mounted on the cam shaft, and built in a heavy substantial manner. The contacts are of the wipe type, are made of tool steel, and are adjustable for wear. A visible indicating spindle shows the amount of each contact.

The entire wearing parts of the timer run in oil, which prevents the burning of the contacts and reduces wear to a minimum.

Wiring from the timer to each spark plug is protected by a five-ampere, enclosed fuse; thus, should any spark plug become damaged or short-circuited, the other plugs would not be affected.

Any spark plug can be removed and replaced while the engine is in operation.

By individual adjustments, the time of ignition in any cylinder end can be set; and by a single handle, the time of ignition in all cylinders can be varied simultaneously. Any of these adjustments can be made equally as well with the engine in operation as otherwise.

Pneumatic Starting Device and Control. The pneumatic starting gear consists of two shifter pistons mounted within the valve lever carrying pins on the rear cylinder. These pistons are shifted by throwing a small three-way cock, which applies compressed air to the one side of the piston, which in turn moves the valve levers, bringing the cam rollers in line with an auxiliary cam. This puts all the valves in both ends of the cylinder into two-cycle action; or in other words, makes a poppet valve air or steam engine out of this cylinder.

This permits the engine to be started on any stroke and on either quarter. Since the operation is entirely automatic, the engine will run as long as compressed air is applied.

The other double-acting cylinder continues to operate as a 4-cycle gas engine, and takes up its explosions after the first revolution. Compressed air is then shut off from the starting cylinder, and the three-way cock reversed. This permits the cam levers to return to their normal position, and that cylinder immediately goes into 4-cycle action.

The starting gear adds practically no complication to the engine, as the regular inlet and exhaust valves are used for distributing air.

There are no extra shafts or auxiliary valves, nor any tappings into the cylinders. There is only one compressed air pipe leading to the sole plate, air being delivered into the fuel duct and entering the cylinders *via* the inlet valves.

On a controlling pedestal is mounted a lever for controlling the free air supply, the three-way cock, the compressed air throttle wheel, the lever for controlling the gas supply, and the ignition switch. This permits the engineer to start the engine, bring it up to speed and adjust the air and gas to a correct mixture without leaving his position.

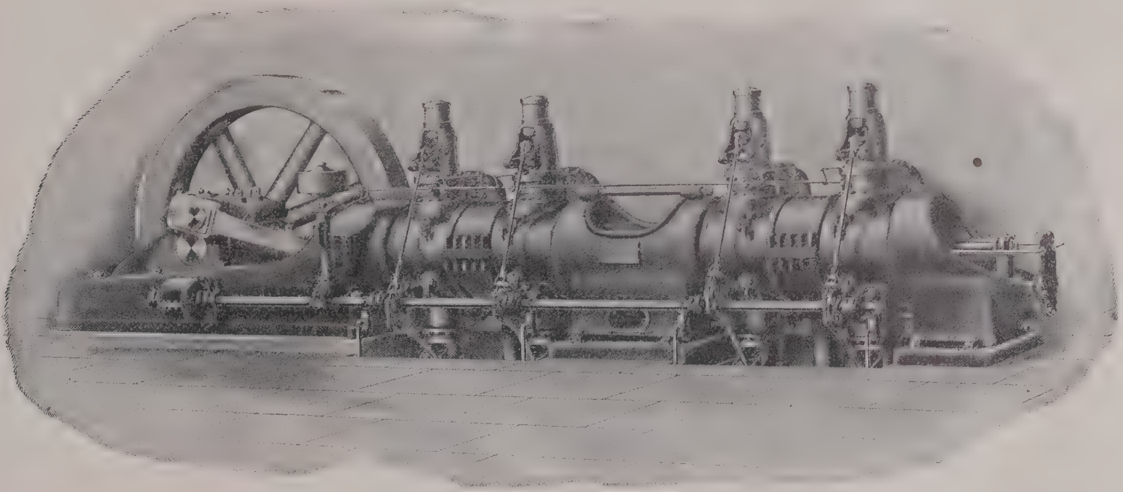


Fig. 627.—4-cycle Double-acting Tandem Engine, Mesta Machine Co.

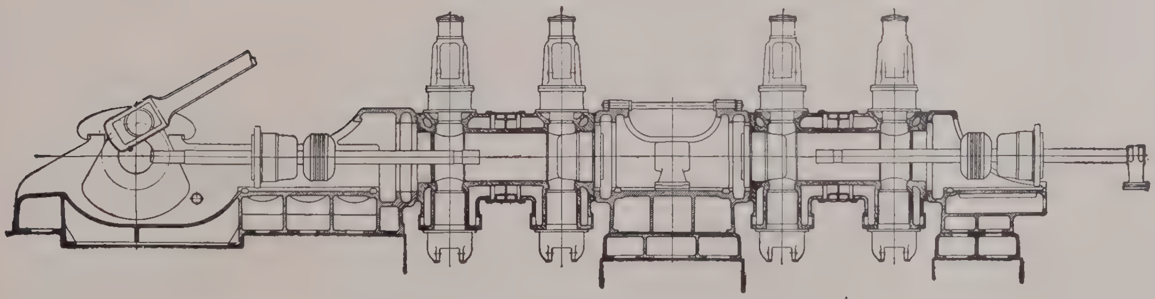


Fig. 628.

7. The Mesta Machine Co., Pittsburg, Pa. This company is at present engaged in the development of a 500 B.H.P. engine, of which Figs. 627 and 628 show the essential features. It is intended finally to build these engines in sizes up to 5000 B.H.P.

SPECIFICATIONS FOR MESTA GAS ENGINE

The Mesta gas engine is a horizontal, double-acting 4-cycle machine built either in single- or twin-tandem units and follows what might be called the standard lines of heavy duty prime movers of this kind, laid down in the well-known Nürnberg type, modified to meet the American requirements.

Frame. The main frame or bed plate is plain and massive in its construction, combining strength and accessibility. Special attention has been given to the rigid connection between the bed plate and forward cylinder obtained by a double flange construction connected by heavy ribs. Following the American practice the crank pin is overhung and the main bearing is designed to meet this arrangement. The bearing is made of four parts with liner adjustment for the forward quarter box and wedge for the rear. This method offers less possibility of improper alignment so often found where both quarter boxes are wedge adjusted. The bearing cap is of the interlocking construction, acting both as compression and tension member over the jaw of the bearing, thus insuring great rigidity.

Cylinders. The cylinders are cast of vanadium steel in order to withstand the high vibratory stresses caused by alternate explosions in the two cylinder ends and also in order to reduce the section of cylinder walls to a minimum, thus increasing the cooling effect of the jacket. The central portion of the jacket is cast open and covered with a split band, which can easily be removed, in case it should be required for any extensive cleaning around the cylinder of accumulated sediment from the cooling water. The lower part of the cylinder is also provided with covers which permit an easy access to the annular space around the exhaust-valve bonnet.

The inlet valve is located on top of the cylinder and the exhaust valve on bottom. This arrangement has the advantage of symmetrical structure, tending to equalize the strains in the cylinder. A further advantage in this arrangement of placing the inlet and exhaust valves as far apart as possible is that the heating of the incoming mixture is diminished. The heating results in decreased unit density of mixture, and therefore means loss in the capacity of the engine. In placing the exhaust valve at the lowest point in the cylinder, dust and foreign matter is more easily swept out than in any side chamber construction.

Valves. Both inlet and exhaust valves are of the mushroom type and operated by means of rolling levers and pull rods actuated by a single eccentric for each two valves. On the inlet valve stem is mounted a mixing valve consisting of a hollow sleeve which closely fits the annular gas and air passage. Ports in this sleeve register with ports in the gas and air passage, when the valve through its rolling lever is opened. The quality and quantity of the incoming mixture is controlled by independent valves in the gas and air passages actuated by the governor according to change of load.

Pistons. An important feature in the design of this piston is its being cast in one piece, with a perfectly symmetrical distribution of metal, thus overcoming to a large extent the liability of shrinkage strains and resulting tendency to crack under service. The water circulation inside the piston is obtained without the use of adjustable parts. There are no cored holes in the outer walls and a smooth surface is presented without metal plugs and their liability to leak. The piston is held floating free from the cylinder wall by means of an intermediate and rear cross-head and the piston rings touching the walls are so designed as to give a uniform bearing pressure.

The Piston Rod is made in halves and connected in the intermediate cross-head. Provision is made for adjustment of the rod both in the horizontal as well as vertical plane. Water is introduced to the rod in the center and discharged at both ends.

Cylinder Heads. These are cast in one piece, cored out for cooling water, and properly recessed for the metallic packing, which is set so far back from the inner end of the head as not to require cooling in addition to that furnished for the head. The construction of the head is such as to allow for changes in compression if desired.

Ignition. Two igniters are placed in each end of the combustion chamber, one at the top and one at the bottom, and are of the magnetic make-and-break type. A timing device controls the spark, which can be advanced or retarded to suit condition. Provision is made for connecting the timer with the governor in order to be able to change the point of ignition for change in load.

Air Starting. The simplest method of starting is obtained by introducing compressed air to the cylinders at a period corresponding to the power stroke in normal operation. This is accomplished by four cam operated poppet valves in the air main to the cylinders which automatically fall out of gear after the air is turned off. Check valves are located in the cylinder at the point of air admission and as soon as the engine gets an explosion the valve will close due to the higher internal pressure. The air is then turned off and the engine is now in operation.

8. The Allis-Chalmers Co., Milwaukee, Wis. This company was originally the American licensee of the Nürnberg engine, but has within the last year or two developed an engine of somewhat different type. The main changes from the Nürnberg design apparently consist in the use of the side-crank frame and in the combining of the main inlet and mixing and regulating valve into one housing. Figs. 629, 630, 631 and 632 will serve to show the important features of the design. The inlet-valve gearing and the regulation being radically different will need special mention, and the following description, in connection with Fig. 633, is taken from the Gas Engine Report above mentioned.

"The operation of gas valve will be readily understood by referring to the figure. At *H* is shown the gas valve. This valve is of the double-seated type and is located concentric with the main inlet-valve stem. The gas enters to the space above this valve from the gas and air manifold *N*. Valve *H* is operated by means of two rods which connect it with cross-head *V*, roller lever *U* and lever pin *S*, which pin is allowed to move up and down with the small cross-head *V*. The fulcrum lever *T* is forked on its inner end, and the ends of this fork are pivoted by pins to the valve bonnet; these pins being stationary in this bonnet and not moving with cross-head.

Rolling lever *U* is connected to main inlet valve rolling lever through the connection as shown, and both levers move in unison. The end of fulcrum lever *T* is connected by means of its rod to the small eccentric placed on the shaft *F*. This shaft is rotated by the governor through the rod *E*, and this causes the end of lever *T* to take a different position dependent on the governor and thus the time of opening and lift of gas valve is changed accordingly.

"It has been found that for gases such as blast furnace and producer gases, a double-seated valve will work freely, even after it has become coated with impurities contained in these gases, and it does not tend to stick, as would any type of cylindrical or piston-valve construction, and due to this it is possible to operate a double-seated valve through the means of a simple governor, and not necessary to complicate the governor arrangement by resorting to the use of a relay of any description.

The small eccentric before mentioned is so arranged that it can be thrown out of gear and held in such position that the gas valve has no lift. This is useful when it is desired to cut off the gas from any of the cylinder ends. After the gas passes the valve *H* it mixes with the air which passes in through the holes arranged in the sleeve extending into air passage; the flow of air is at nearly right angles to the current of inflowing gases, thus obtaining thorough mixing of gas and air on their passage to the cylinder. The air is proportioned by a valve located in air and gas manifold *N*; this valve being operated through means of the hand-wheel shown at *D*, and, when once set, does not need changing unless there is great variation in the quality of the gas supplied."

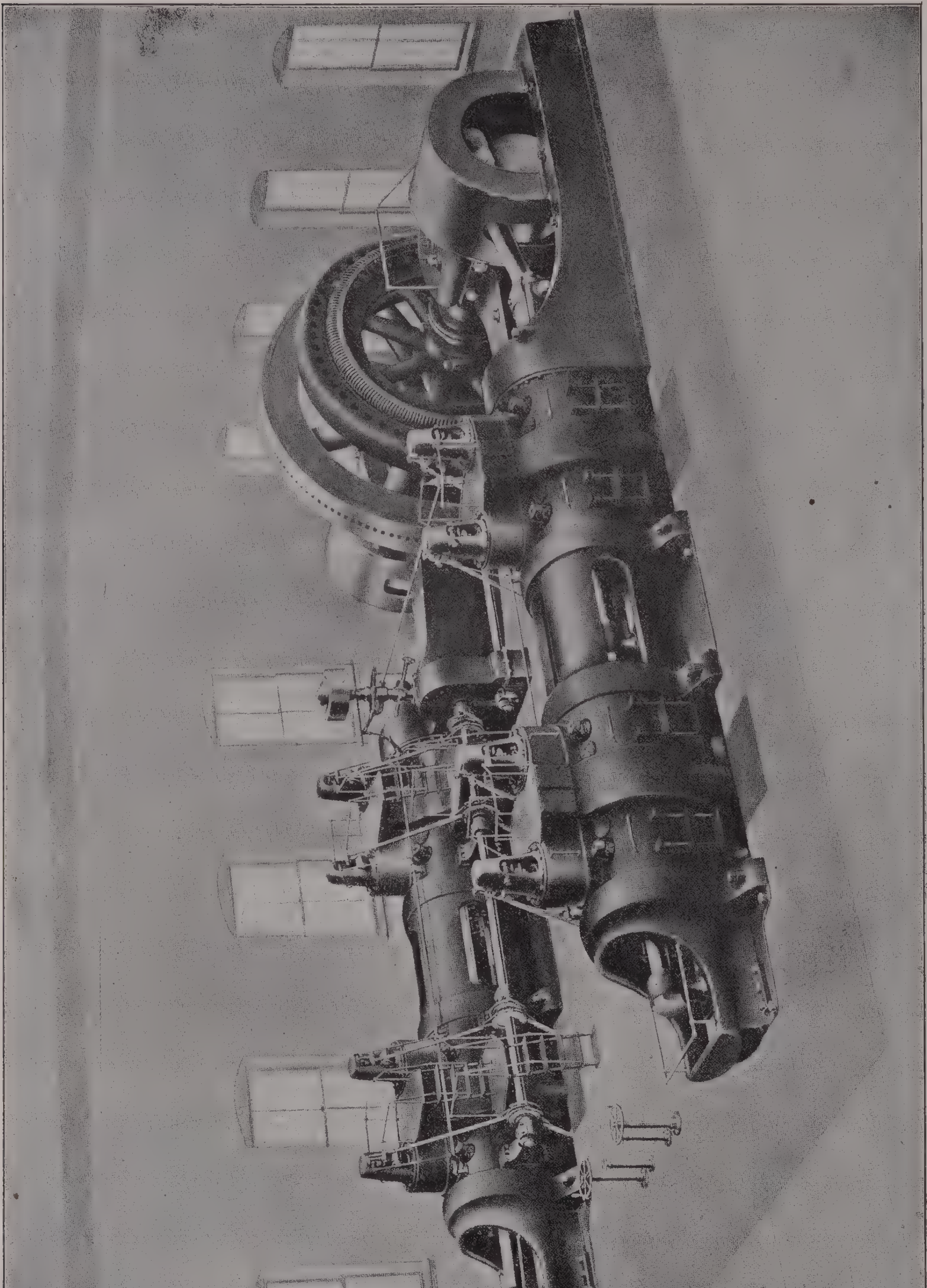


FIG. 629.—Allis-Chalmers 4-cycle Double-acting Twin Tandem Engine.

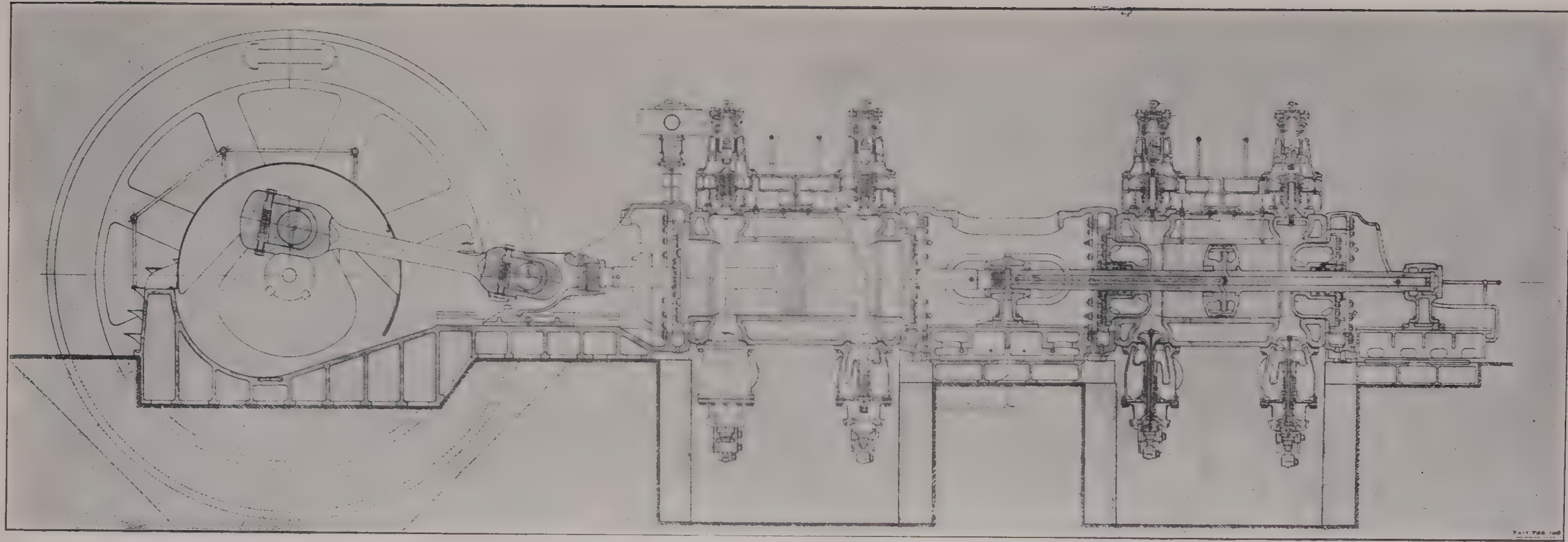


FIG. 630.—Longitudinal Cross-section, Allis-Chalmers Engine.

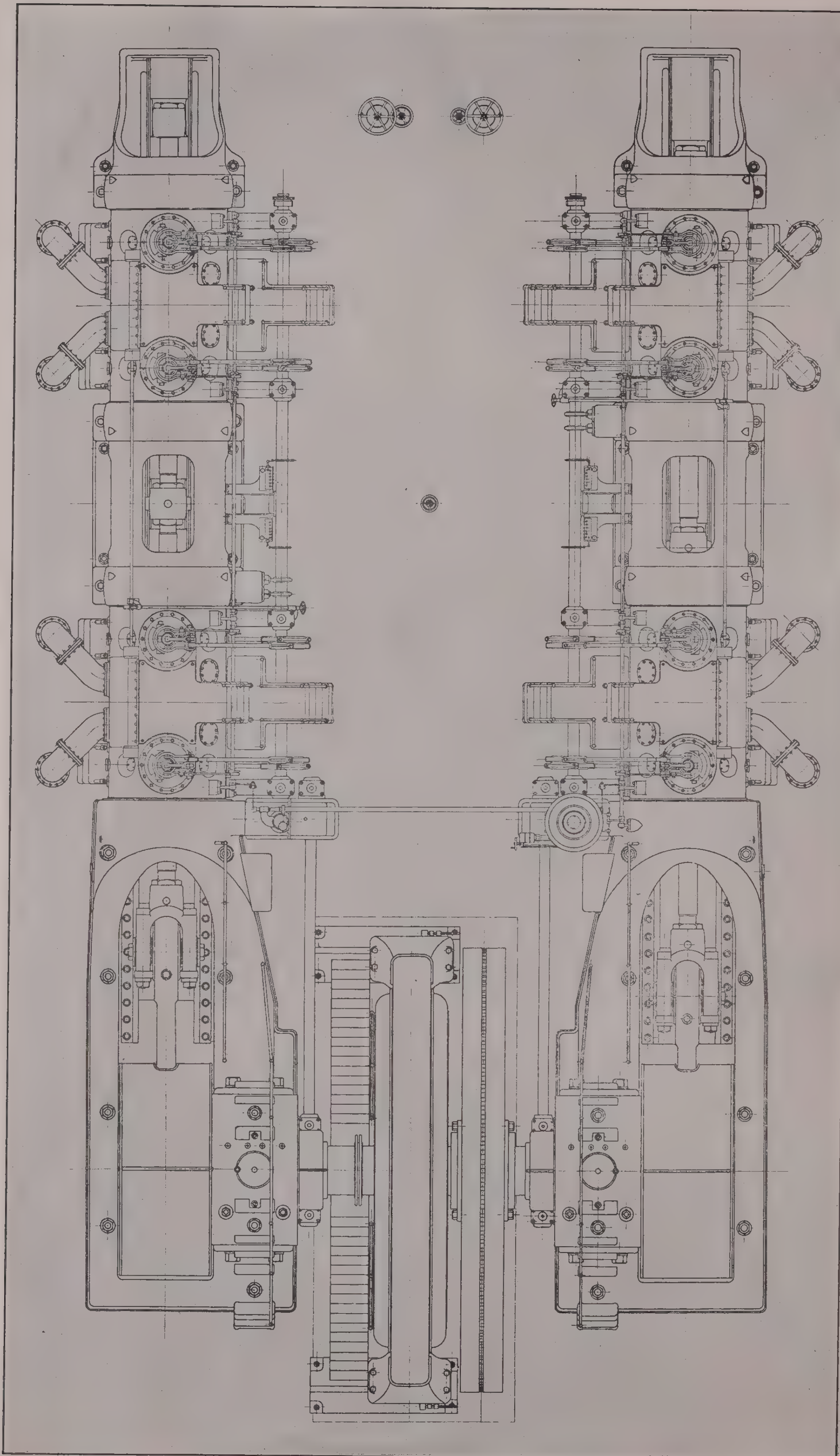


FIG. 631.—Plan View, Allis-Chalmers Engine.

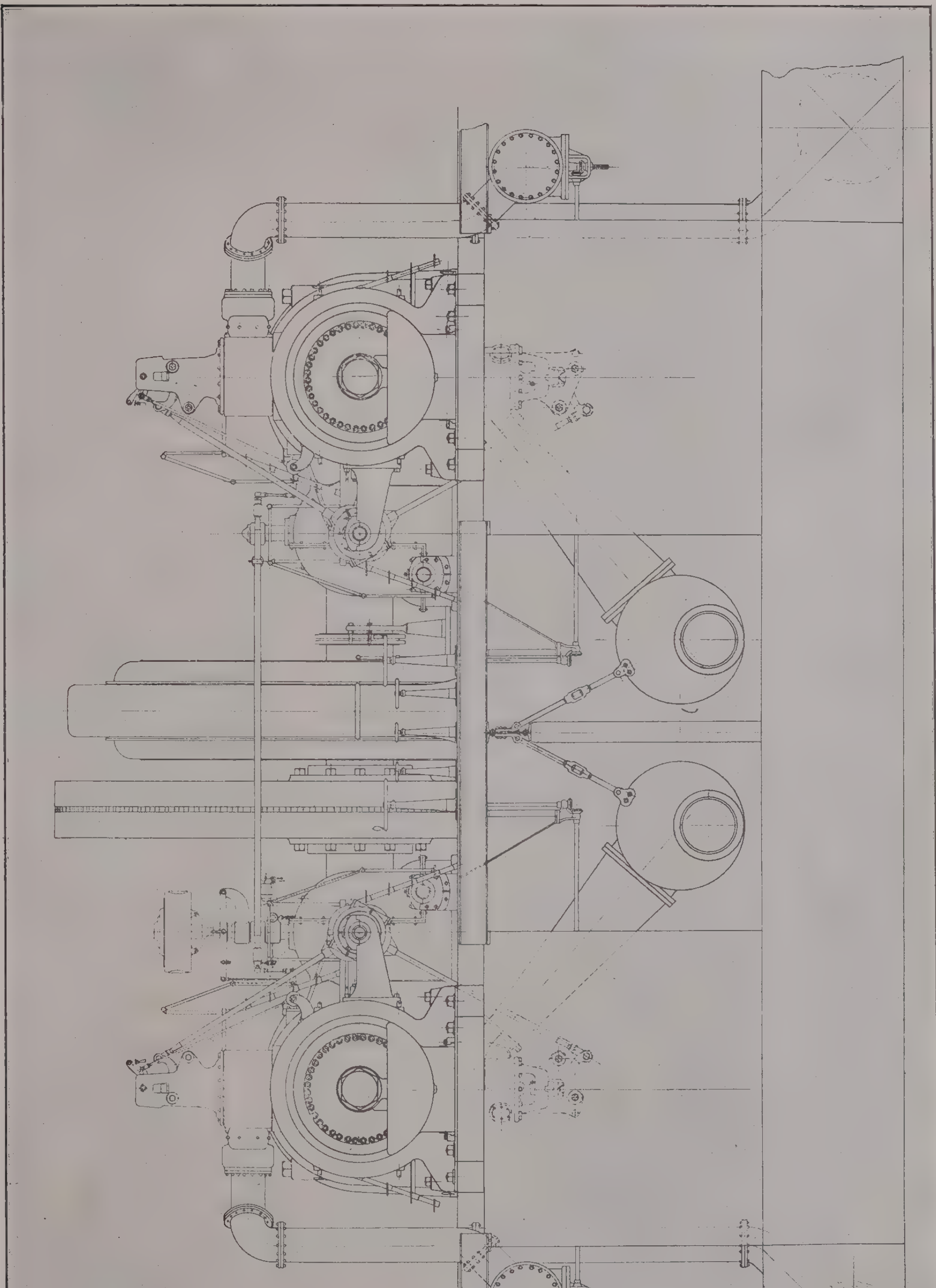


FIG. 632.—End View, Allis-Chalmers Engine.

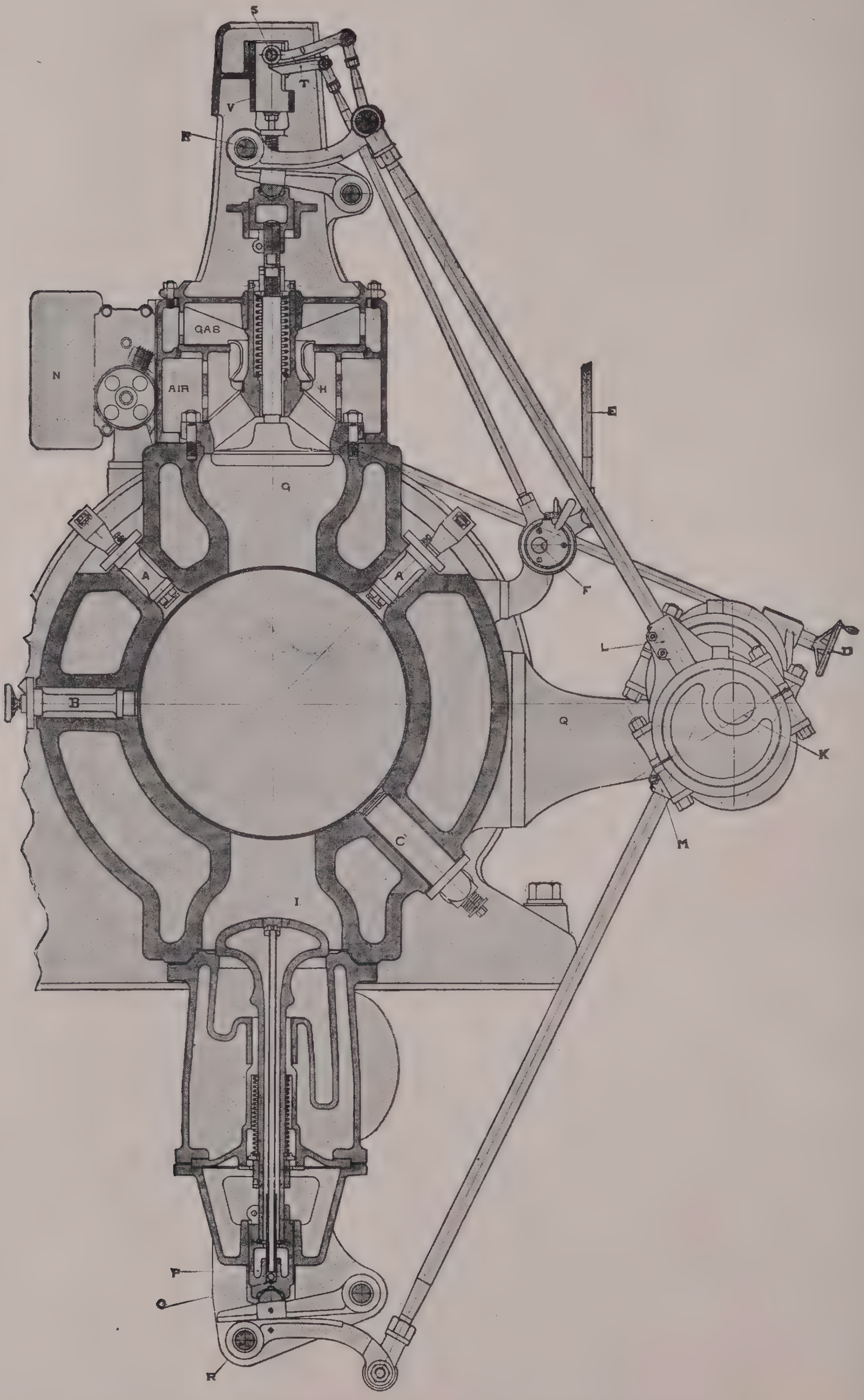


FIG. 633.—Cross-section of Cylinder and Valves, Allis-Chalmers Engine.

9. The William Tod Co., Youngstown, Ohio. The engines built by this company, like all of the large American engines with the exception of the De la Vergne-Körting, is of the 4-cycle double-acting tandem type. Figs. 634, 635, and 636 show the general make-up of a 30×42″ engine, which has been in service since March, 1908. A 42×60″ engine direct-connected to two 80×60″ blowing tubs is now building.

The three general views above mentioned, together with the cross-sectional cuts in Fig. 637, will serve to explain the construction. The use of the side crank frame and the split cylinder jacket has about become standard practice. The cylinders are cut in half, bolted at the middle, and are supported at the ends only, the two cylinders, the distance piece, the tail frame, and the main frame all being held together by four strong continuous tie-rods.

The valves are located top and bottom. For one cylinder-end they are operated by one eccentric, the exhaust valve direct, the inlet valve by means of a bell crank. There is nothing out of the ordinary in the construction of either exhaust or main inlet valve, but the mixing and governing arrangements are different from those found in any other American engine.

The engine governs by proportioning the amount of gas to the load, there being practically constant compression. The mixing valve is a three-seated sleeve shown in Fig. 638 just above and concentric with the main inlet valve. This valve moves up and down with the inlet valve. Above this mixing valve is placed the regulating

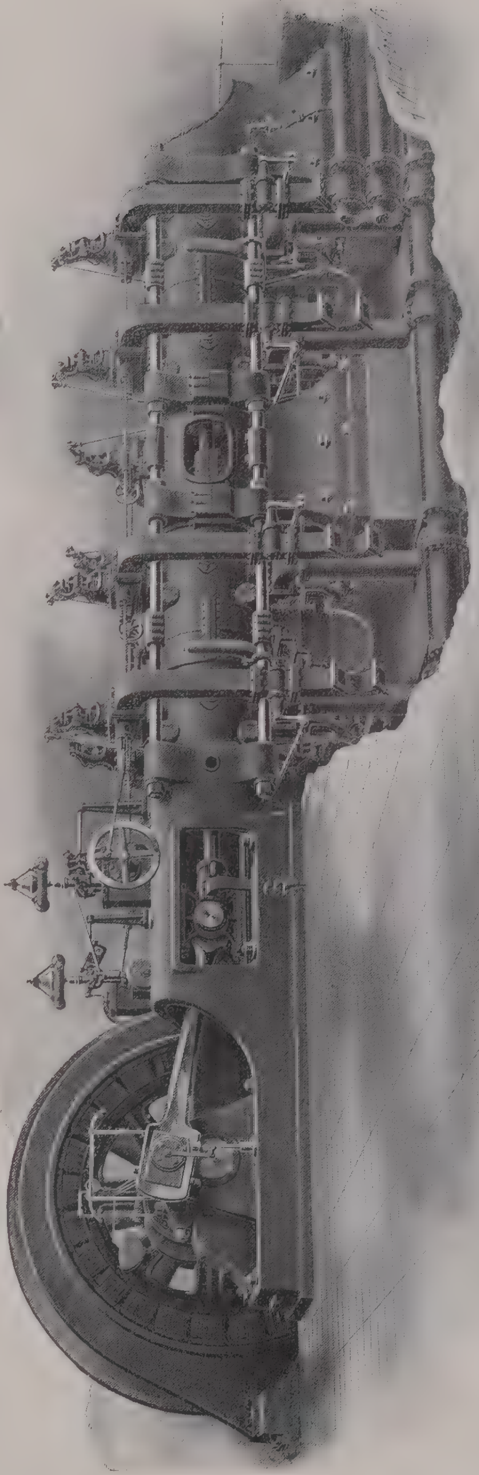


FIG. 634.—Double-acting Tandem 4-cycle Engine, Wm. Tod Co.

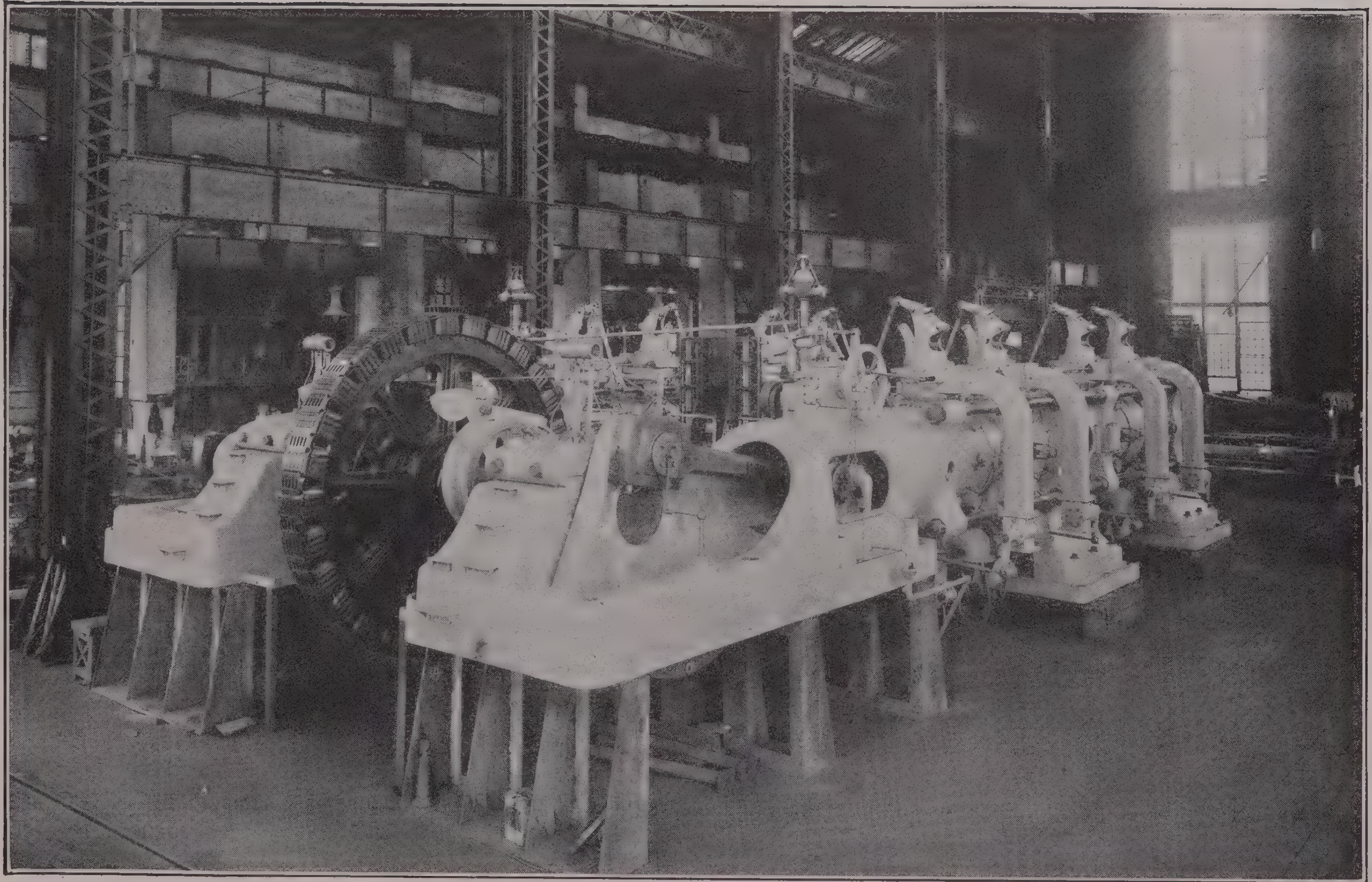


FIG. 635.—Tod Double-acting Tandem Engine.

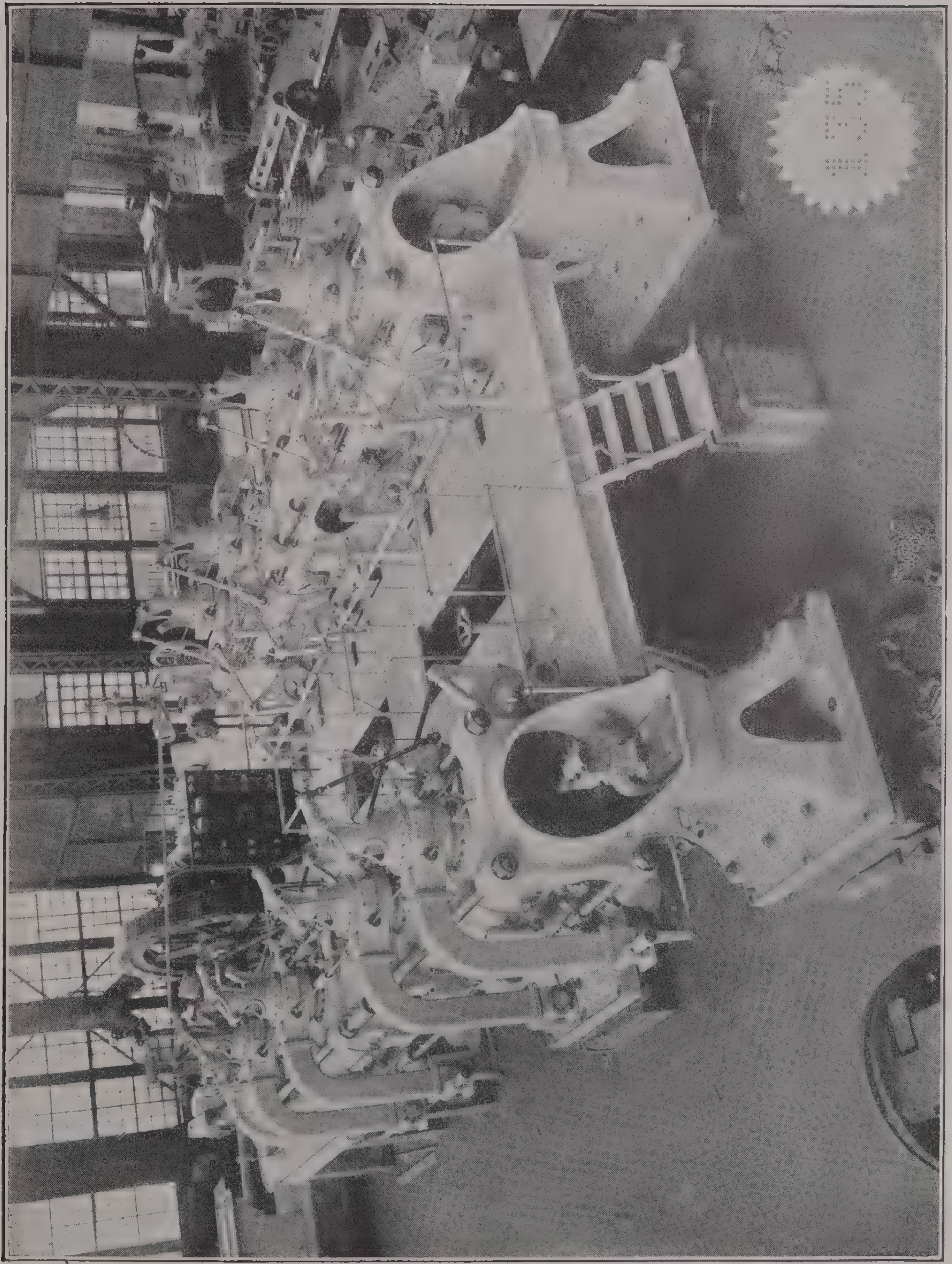


Fig. 636.—Tod Double-acting Tandem Engine.

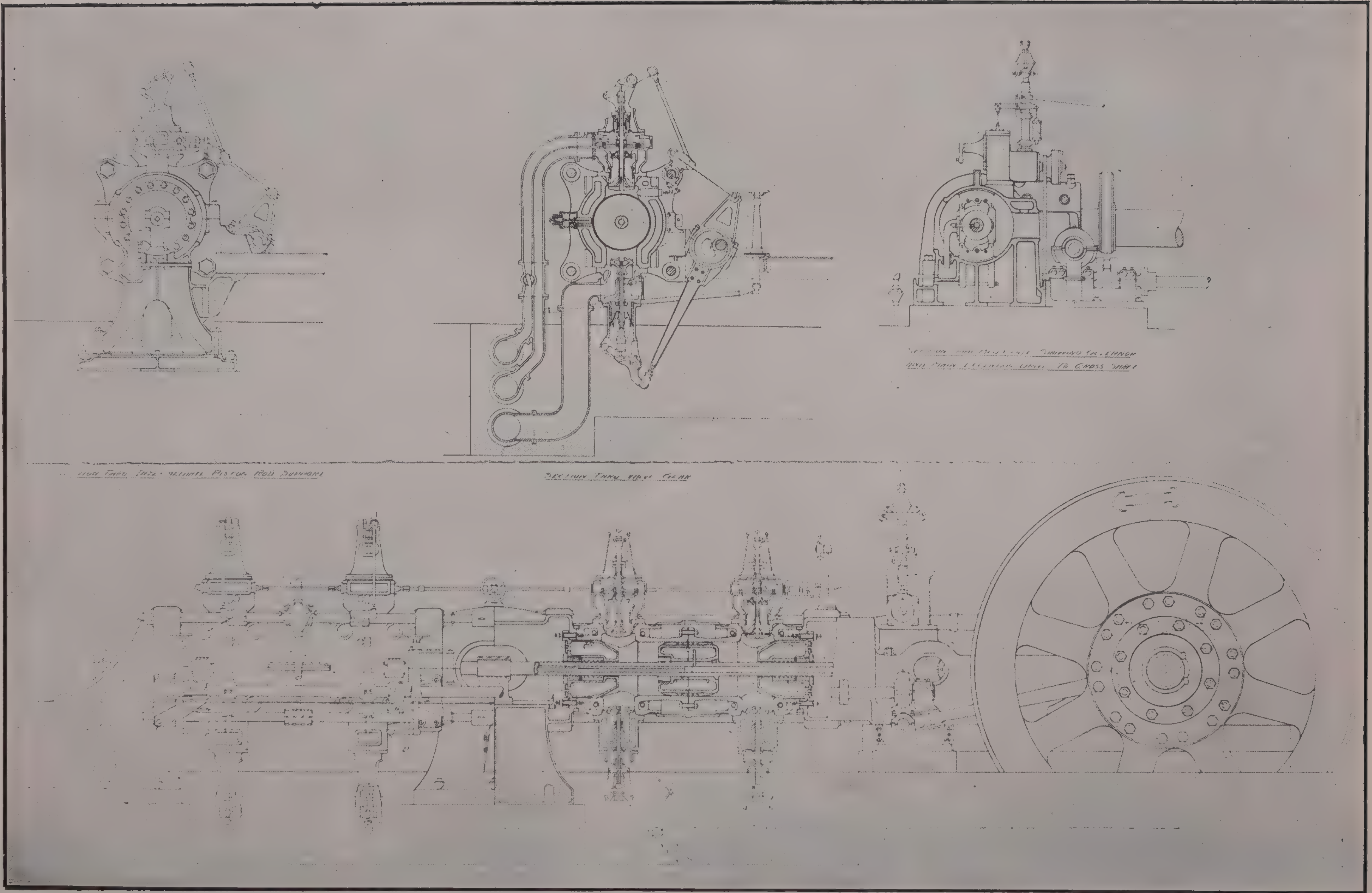


FIG. 637.—Cross-sections Tod Double-acting Tandem 4-cycle Engine

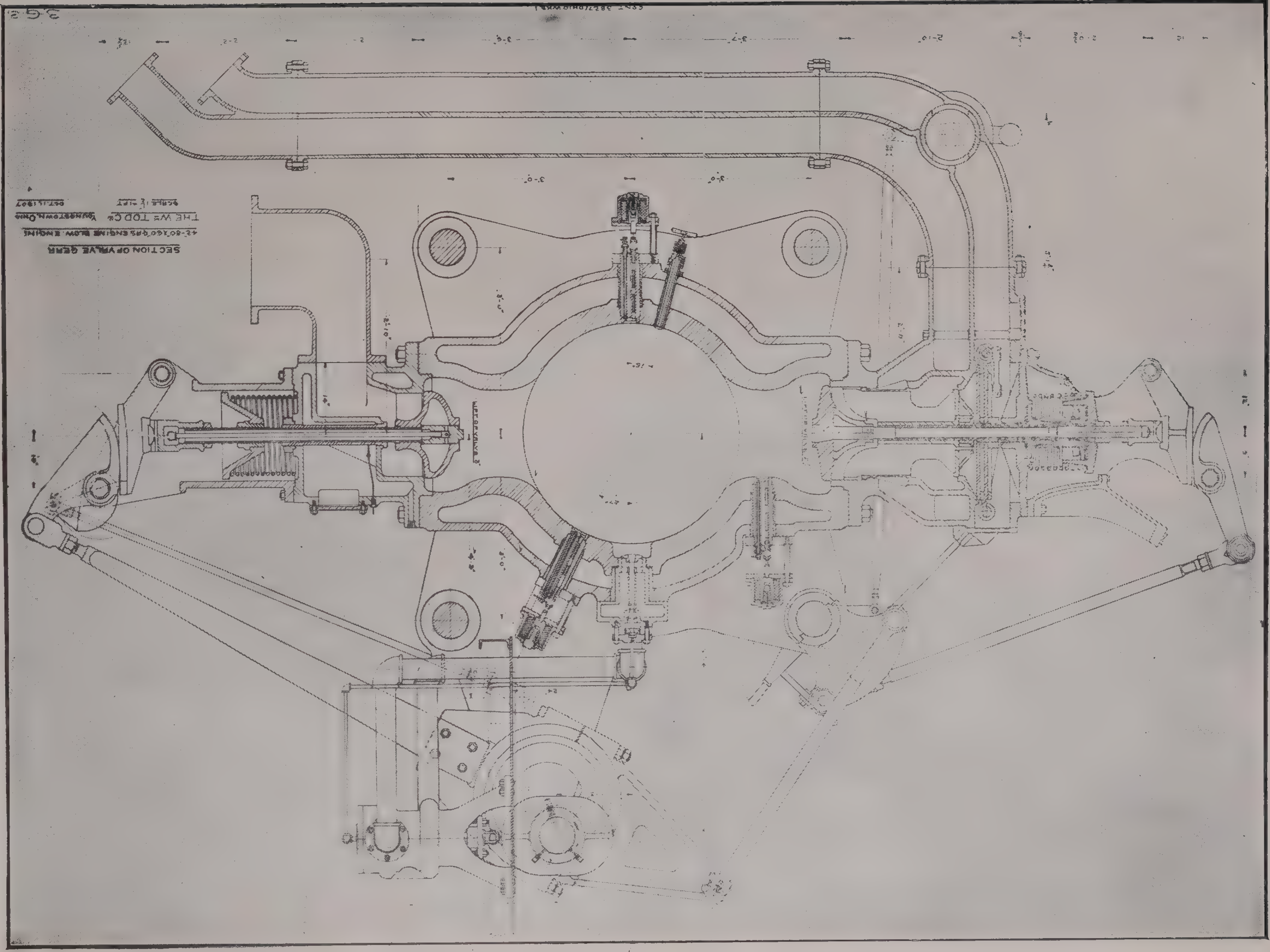


FIG. 638.—Cross-section of Cylinder and Valves, Tod Double-acting Tandem Engine.

valve of which a plan view is shown in Fig. 639. This valve consists of four disks having segmental slots as shown. The upper disk is oscillated about its center by means of the longitudinal rod *H* which passes along both cylinders and operates the disks in the other valve housing in exactly the same way. Owing to the motion, the slots in the disk alternately open and close similar slots in the stationary disk directly below it. Rod *H* receives its constant back-and-forth stroke from a small crank which is clearly shown above the cross-head guide in Figs. 634 and 635. At full load the motion of the small crank is in phase with the motion of the main crank, that is, the disk valve opens at the beginning of the suction stroke, and furnishes gas full stroke. As the load drops, the governor, by means of the floating spur gears shown in Fig. 640, causes the governor crank to lag behind the main crank, thus retarding the time of opening of the gas valve, the cylinder drawing only air during the first part of the stroke.

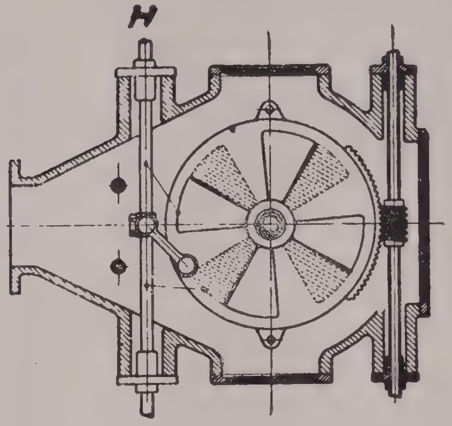


Fig. 639.—Governing Valve, Tod Engine.

This method of governing has been found entirely successful in practice as long as the quality of gas was fairly constant or did not vary too suddenly. To take care of a change in the quality of gas, there is placed under the stationary disk a disk throttle whose position may be changed by hand through a worm, thus adjusting the gas to air according to the quality of the former.

No operating results from this engine are available. Although the heating value of the gas varied suddenly from 70 to 110 B.T.U., which caused some irregularity in firing, the engine regulated satisfactorily and carried overloads up to 20% for long periods.

10. The A. H. Alberger Co., Buffalo, N. Y. This firm builds a type of engine which is finding much application for medium powers, that is, the single-acting tandem. The main features of the construction become plain from Figs. 641, 642 and 643. The design has the merit of great simplicity. An unusual feature in the cylinder construction is found in the use of an unjacketed head for the back cylinder. The valves are placed side by side in a chamber at the side of the cylinder, as shown in Figs. 641 and 644. Neither valve is water-cooled and both are mechanically operated from a lay shaft underneath by means of simple cams. A small eccentric on the lay shaft, Fig. 641, also operates the make-and-break igniter at the inlet side of each valve chamber. The two inlet valve chambers are connected by a cast iron header, at the middle of which is mounted the mixing and governing valve, see Fig. 645. This valve consists of a hollow cylinder divided into two parts by a partition across it. Ports cut in the side of the cylinder match with similar ports in the side of the cage surrounding the valve. Gas is admitted to the header through the upper ports of valve and cage, and air through the lower. The vertical adjustment of the valve in the cage controls the ratio of air to gas. Raising the valve by means of a thumb screw located on the outside of the mixing chamber decreases the gas ports but increases the air ports by the same amount, so that the total effective port area remains the same. Lowering the valve has the opposite effect. The mixture is admitted to the header by oscillating the valve in its cage, alternately opening and closing the ports first for one cylinder and

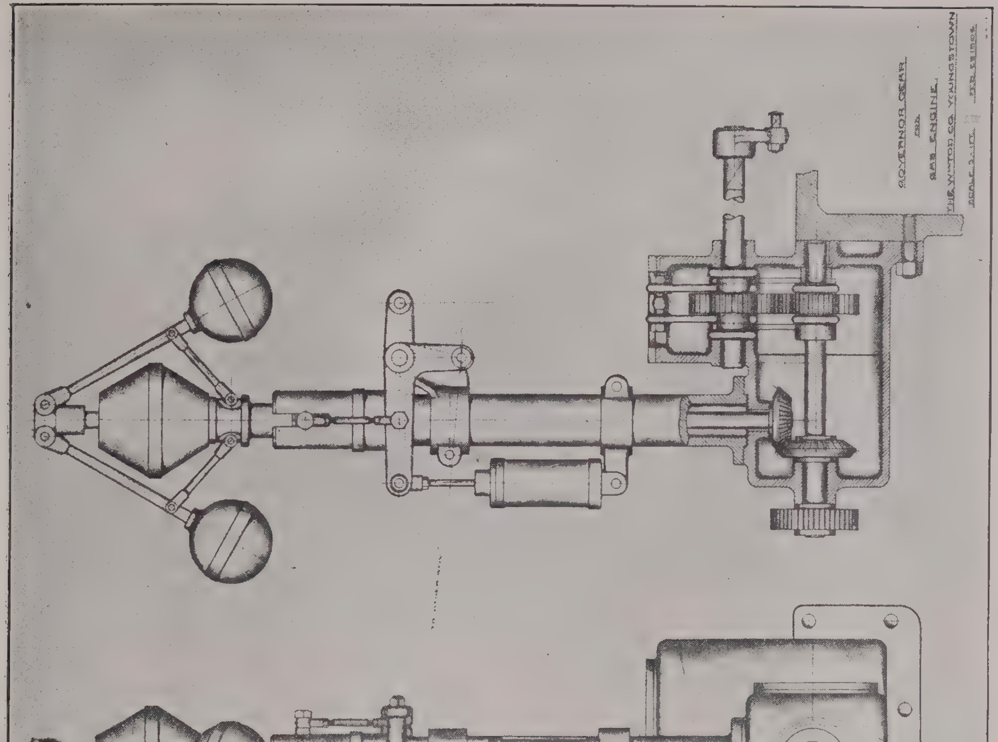


Fig. 640.—Governor, Tod Engine.

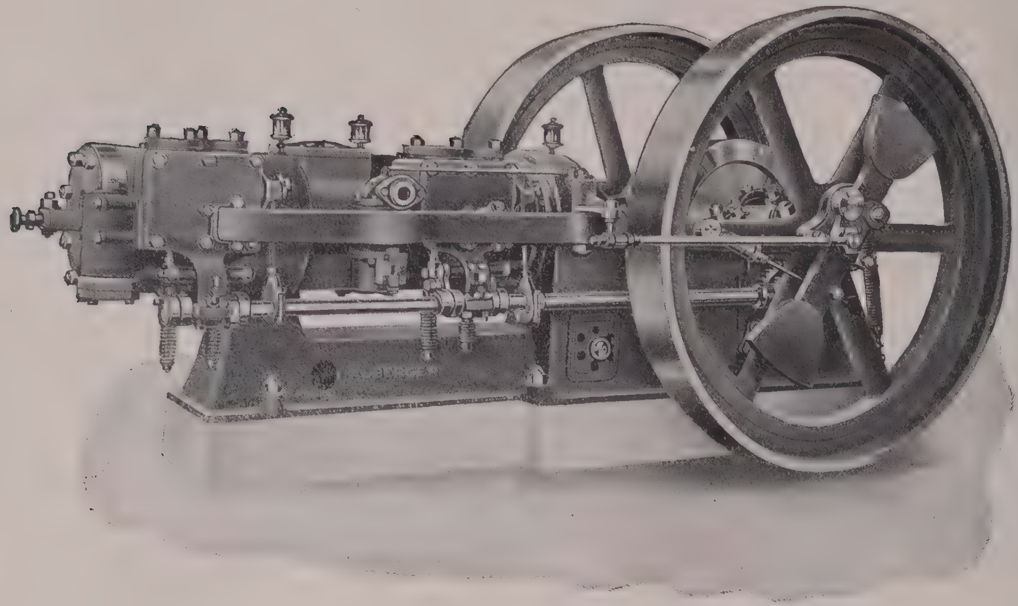


Fig. 641.—Alberger Single-acting Tandem Engine.

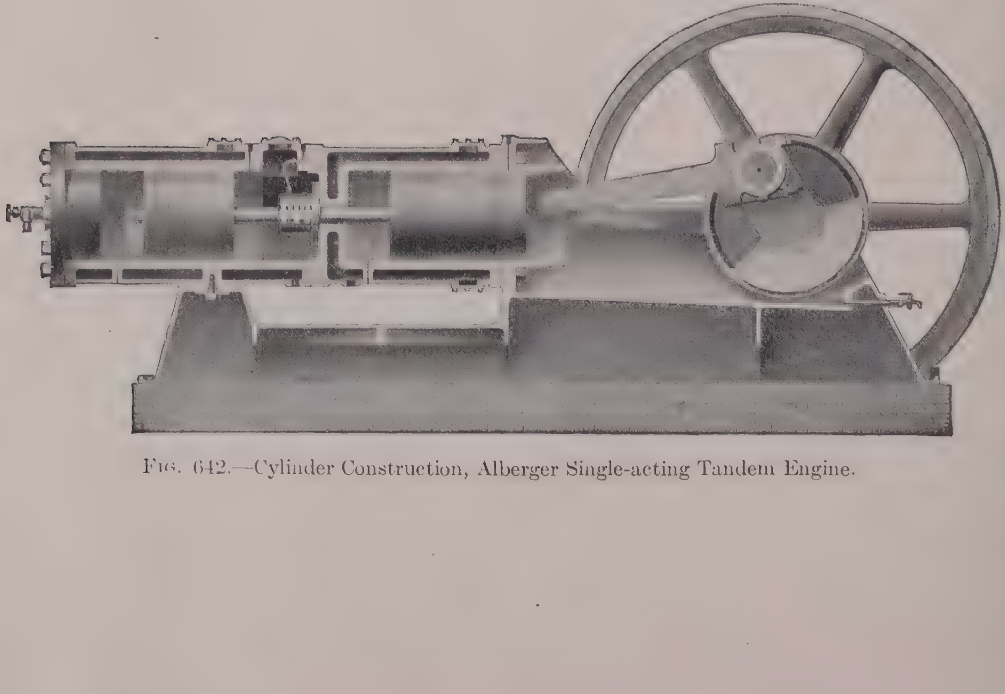


Fig. 642.—Cylinder Construction, Alberger Single-acting Tandem Engine.

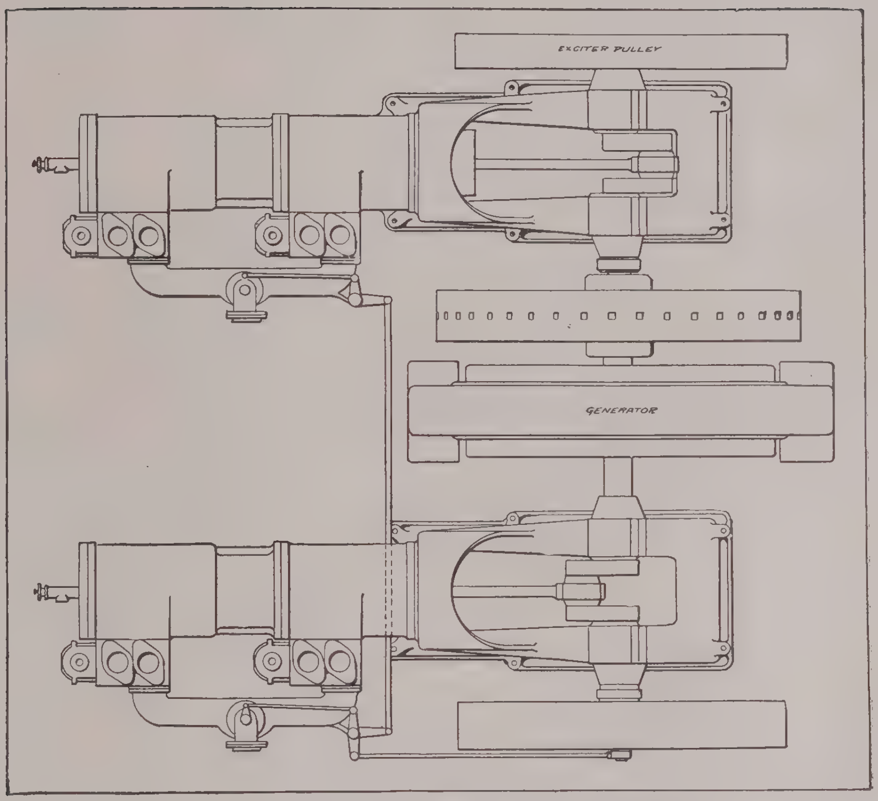


FIG. 643.—Twin-unit Alberger Single-acting Tandem Engine.

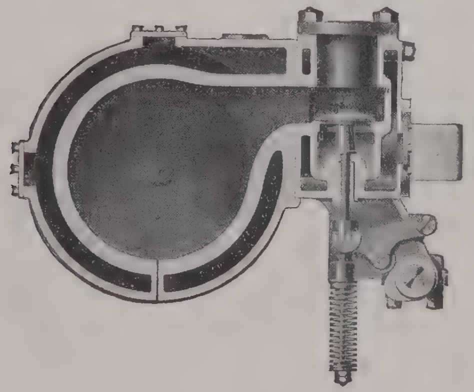


FIG. 644.—Valve Chest, Alberger Engine.

then for the other. This motion is produced by connection to the pin of a Rites inertia governor shown in the wheel in Fig. 641. The position of the governor cuts the mixture off earlier or later during the suction stroke. The use of a governor of this type is unique in gas engine practice. The regulation with this system of governing can be made very close, but, as applied here, it is not possible to change speed while the engine is running. This is desirable sometimes in water-works engines, in engines driving air, gas, or ammonia compressors, etc., for which reason the firm also uses a throttling (fly-ball) governor, operated from the lay shaft, as shown in Fig. 646.

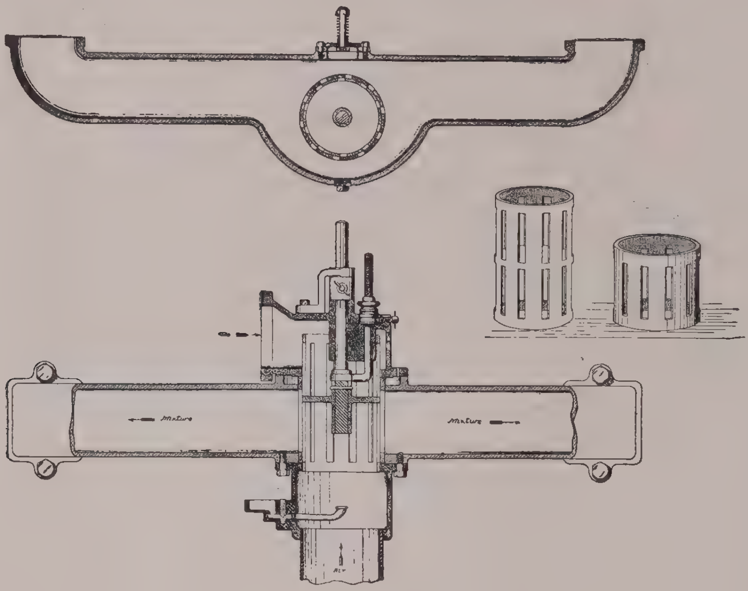


FIG. 645.—Governing Valve, Alberger Engine.

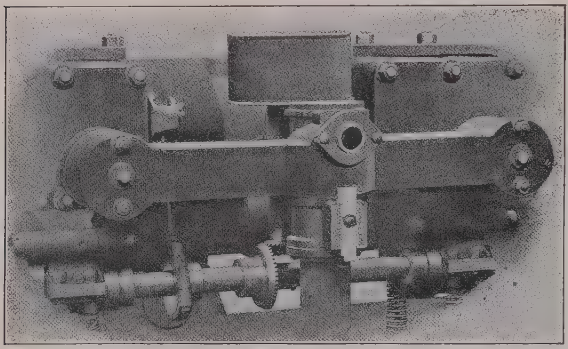


FIG. 646.—Throttling Governor, Alberger Tandem Engine.

Alberger (Buffalo) tandem gas engines are built in sizes from 35–250 B.H.P. for producer gas and from 45–300 B.H.P. for natural gas. Twin tandem units, the layout of one of which is shown in Fig. 643, double these capacities.

11. The Struthers-Wells Co., Warren, Pa. Figs. 647, 648 and 649 show a type of tandem engine, built by this company, which is distinctly different from the one just described. The illustrations are sufficiently clear to explain the construction without further description. The construction of the frame, cylinder heads, and barrels, valve cages, etc., with a view to accessibility and easy removal, is somewhat out of the ordinary and deserves special mention.

This firm also builds several other types of engines, both vertical and horizontal, but information regarding economy, etc., did not seem to be available either for these or for the engine above illustrated. The Report of the National Electric Light Association states that the firm guarantees for the tandem engine a gas consumption of not

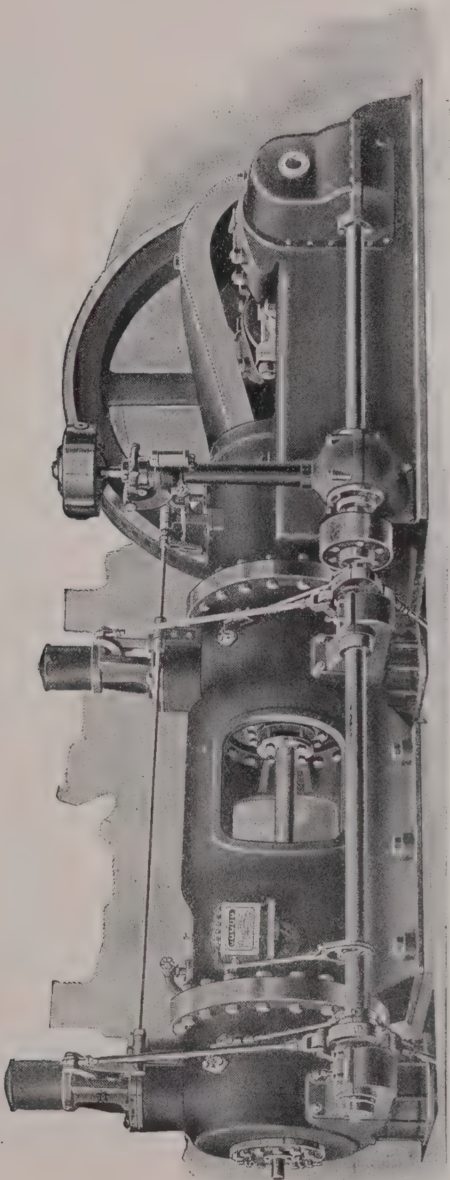


FIG. 647.—Struthers-Wells Single-acting Tandem Engine.

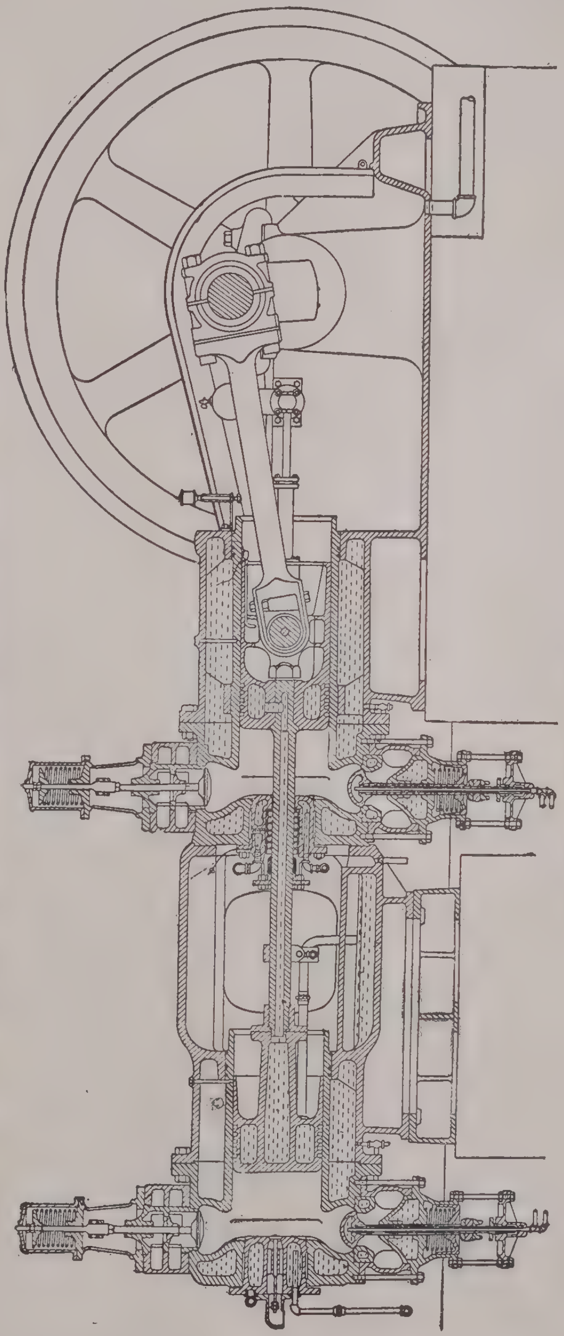


FIG. 648.—Cross-section, Single-acting Tandem Engine.

over 10 cu.ft. per B.H.P. hour for natural gas of approximately 1000 B.T.U. per cu.ft. It is also stated that on a recent test the consumption was shown to be 8.1 cu.ft. per B.H.P. hour, the gas containing a little less than 1000 B.T.U. per cu.ft.

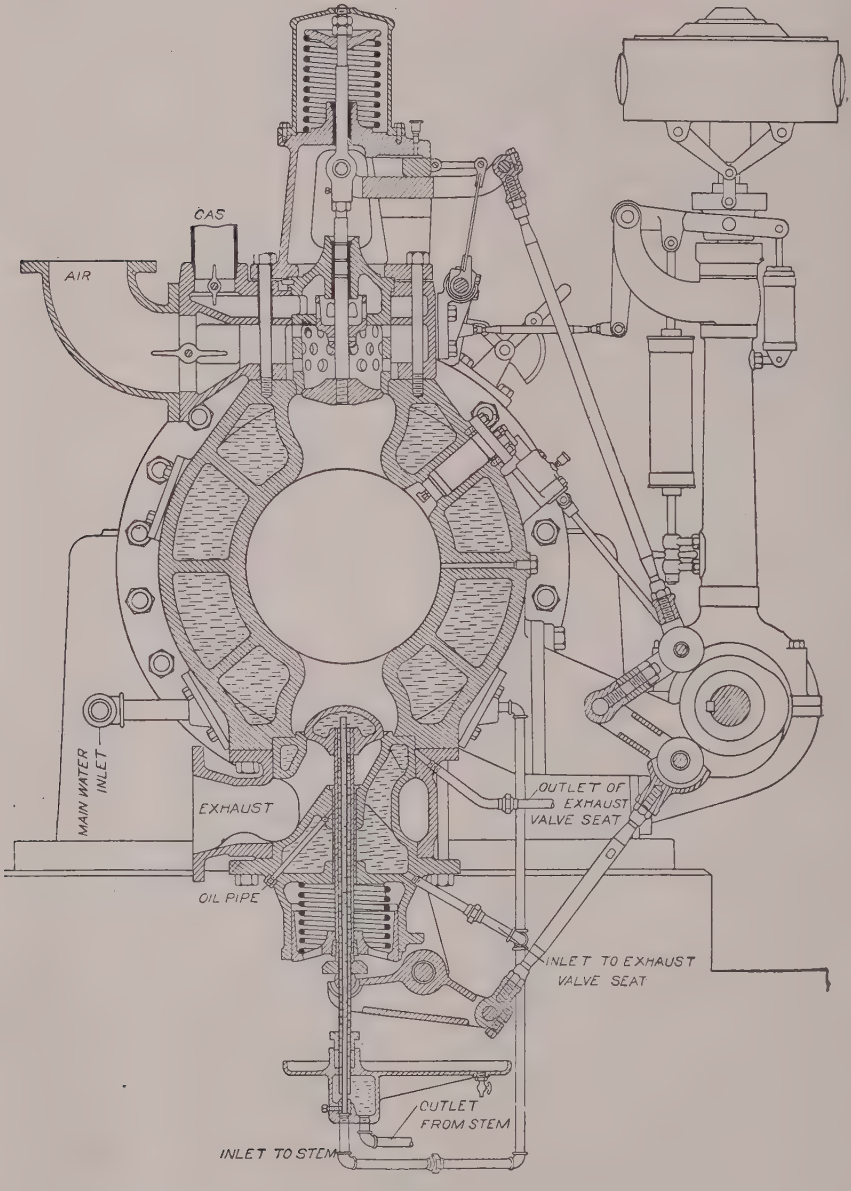


Fig. 649.—Cross-section of Cylinder and Valves, Struthers-Wells Single-acting Tandem Engine.

B. PORTABLE AND SELF-PROPELLED ENGINES

Under this head are considered simple portable power trucks, traction engines, automobile and marine engines. The various types built are nearly all operated with liquid fuel (gasoline, kerosene, and lately also alcohol), mainly because these fuels have the greatest heating value per unit of volume.[1] An exception to this rule is found only in gas traction engines or gas railways, in which the fuel consists of a comparatively small quantity of illuminating gas highly compressed (see p. 508).

I. Portable Engines

Fundamentally our present day portable oil engines are in reality nothing more than engines mounted on trucks, together with muffler, oil tank, water tank, and other auxiliaries. A construction which unites into one harmonious whole all of these component parts, such as is exhibited by portable steam engines, has not yet appeared and is in any case not as easy to obtain as in steam locomobiles, the main part of which consists of the boiler, which in turn may be made to carry the auxiliaries. The oil locomobile, on the other hand, requires a special frame to act as foundation for the engine and the only thing remaining to the designer then is to place the auxiliaries as best he may to obtain good appearance.

The most important market for the portable oil engine is of course created by the agricultural demand, and builders have naturally paid most attention to the needs and requirements of this industry. The builder usually bases the power capacity of the locomobile under consideration upon the nominal power demand of his threshing machine, which requires most power among agricultural machines. Since the internal-combustion engine lacks the overload capacity that would be possessed by a steam locomobile, to be on the safe side the oil traction engine should be given a maximum capacity at least 40 or 50% above the normal power demand of the thresher. Failure to take this important point into account is always followed by trouble for both buyer and builder.

The threshing drums usually make from 1200 to 1400 revolutions per minute. The diameter of the pulleys is from 6½ to 8″ with a width of face of about 8″. The belt speed is in the neighborhood of 40 ft. per sec., so that this as well as the ratio between driver and driven pulley is comparatively high. For good operation the distance between shafts should not be less than 15 ft., and the heavy belt not only brings considerable stress to bear on the crank but also renders difficult the starting of the larger sizes. This trouble is often avoided by the use of a countershaft which drives the drum, so that the engine may be started with the main driving belt at rest. The same end is reached by simpler means by use of any of the well known designs of friction clutch. The belt is to be placed in such a way as not to interfere with the attendance of either engine or thresher. The fuel and water supply carried should last at least ten operating hours.

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[1] A steam locomobile for each effective B.H.P. requires at least 6½ lbs. of coal and about 45 lbs. of water, together, say, 50 lbs. of material. An oil locomobile requires only from about .7 to .9 lb. of oil for fuel, and, if cooling by vaporization is used, about 2.2 lbs. of water per B.H.P. hour. The weight of these materials therefore that must be carried per H.P. hour in the oil engine is therefore only from $\frac{1}{15}$ to $\frac{1}{20}$ of that required by the steam locomobile.

The fuel consumption of locomobiles, as in general that of all portable and self-propelled motors, is usually somewhat greater than that of similar stationary engines of the same capacity. The main reasons for this are found in unavoidable jars and vibrations to which the vehicle is subject and which absorb a certain amount of the power developed, and the less perfect formation of mixture, cooling. etc.

Designs of Locomobiles:

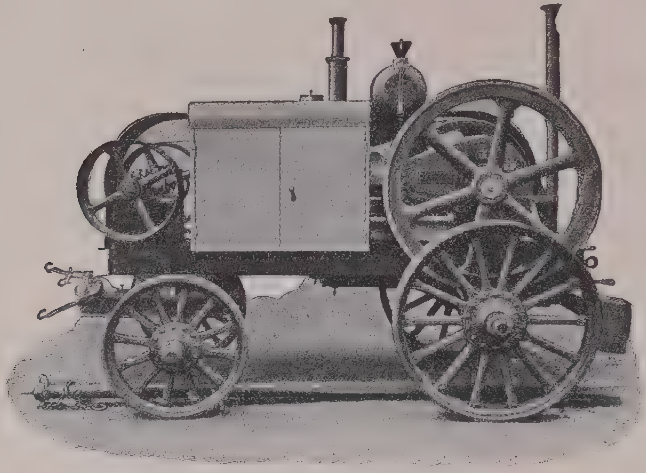


FIG. 650.—Deutz Alcohol Locomobile.

The portable alcohol engine of the Gasmotorenfabrik Deutz, during the last tests under the auspices of the German Agricultural Society, showed the best results. The fuel consumption varied as follows:

Alcohol per B.H.P. hr., lbs., .802 at 16.8 H.P., maximum load.
.855 at 12.1 H.P., normal load.
1.113 at 6.27 H.P., half load.

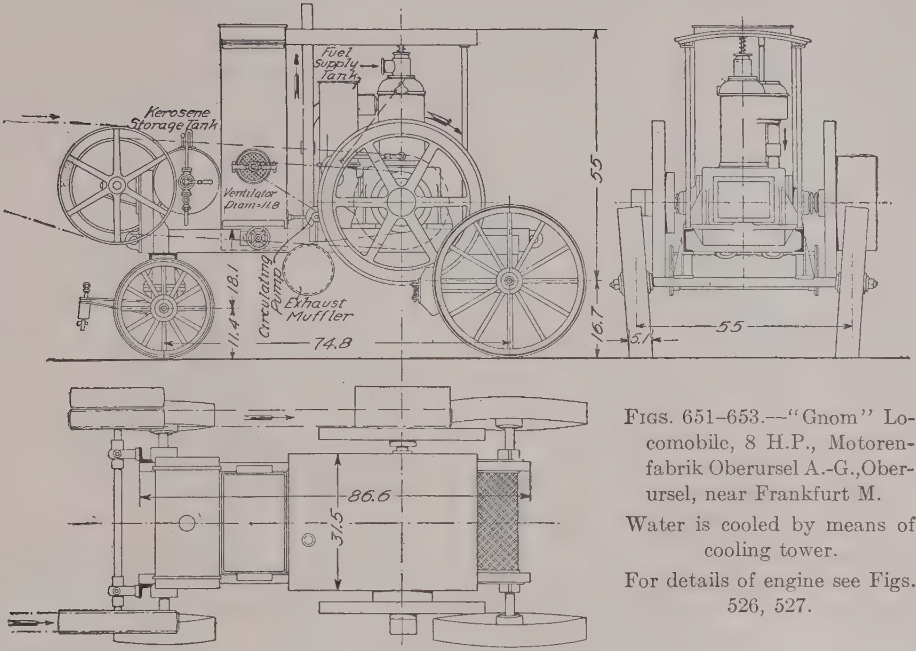
and 4.62 lbs. per hour at no load.

Table 106 shows the dimensions of the usual sizes of locomobiles built by this company.

TABLE 106

DIMENSIONS AND WEIGHTS OF DEUTZ ALCOHOL LOCOMOBILES

Normal rating, B.H.P	4	6	8	12	16	20
R.p.m. of driver on countershaft	450	445	420	420	375	345
Approximate weight, lbs.	4620	5720	6820	8800	11450	14300
Maximum length, without tongue, ft	9.85	10.18	10.50	11.15	12.45	13.10
Maximum width, ft	4.75	5.08	5.42	6.08	6.72	7.20
Maximum height, ft.	6.90	7.05	7.20	7.85	8.53	9.00
Diameter driving wheel, in.	21.3	21.3	23.7	23.7	26.0	28.4
Face, driving wheel, in.	5.5	6.3	7.1	7.9	8.7	9.5
Wheel base, in.	59.0	63.0	67.0	73.0	82.7	89.5
Wheel tread, in.	36.6	39.8	42.9	48.5	52.3	57.5
Width of tire, in.	4.72	5.12	5.51	5.92	6.31	6.70
Diameter front wheels, in.	28.2	29.5	29.5	31.6	35.4	35.4
Diameter rear wheels, in.	39.4	39.4	39.4	43.4	47.3	47.3



Figs. 651–653.—"Gnom" Locomobile, 8 H.P., Motorenfabrik Oberursel A.-G., Oberursel, near Frankfurt M.

Water is cooled by means of cooling tower.

For details of engine see Figs. 526, 527.

Table 107

DIMENSIONS AND WEIGHTS OF GNOM LOCOMOBILES

Normal rating, B.H.P.	2	3	4	5	6	8	10	12	15	20
R.p.m. of countershaft	360	350	300	300	250	240	250	250	250	250
Weight including counter,* lbs.	....	....	....	....	....	....	....	....	....	....
Diameter of countershaft or driving pulley, in.	19.8	19.8	19.8	23.6	35.5	35.5	39.4	39.4	39.4	39.4
Width of countershaft or driving pulley, in.	5.5	6.3	8.6	8.6	5.5	5.9	7.5	7.5	7.5	9.1
Width of belt, in.	2.36	2.76	3.53	3.94	4.72	5.12	5.90	6.70	6.70	7.90

* Counter-shafts used above 6 H.P. The driving pulley or the counter-shaft runs a trifle slower than the pulley on the crank-shaft (see figures in Table 62, p. 361).

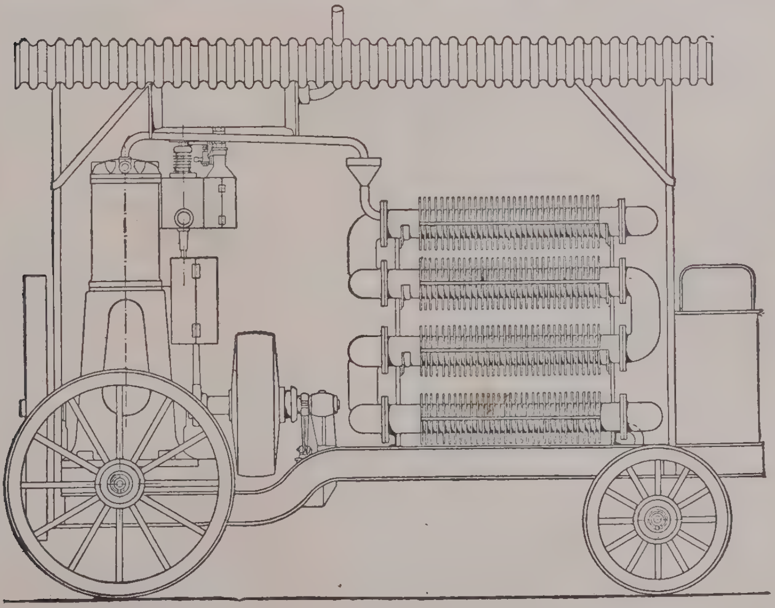
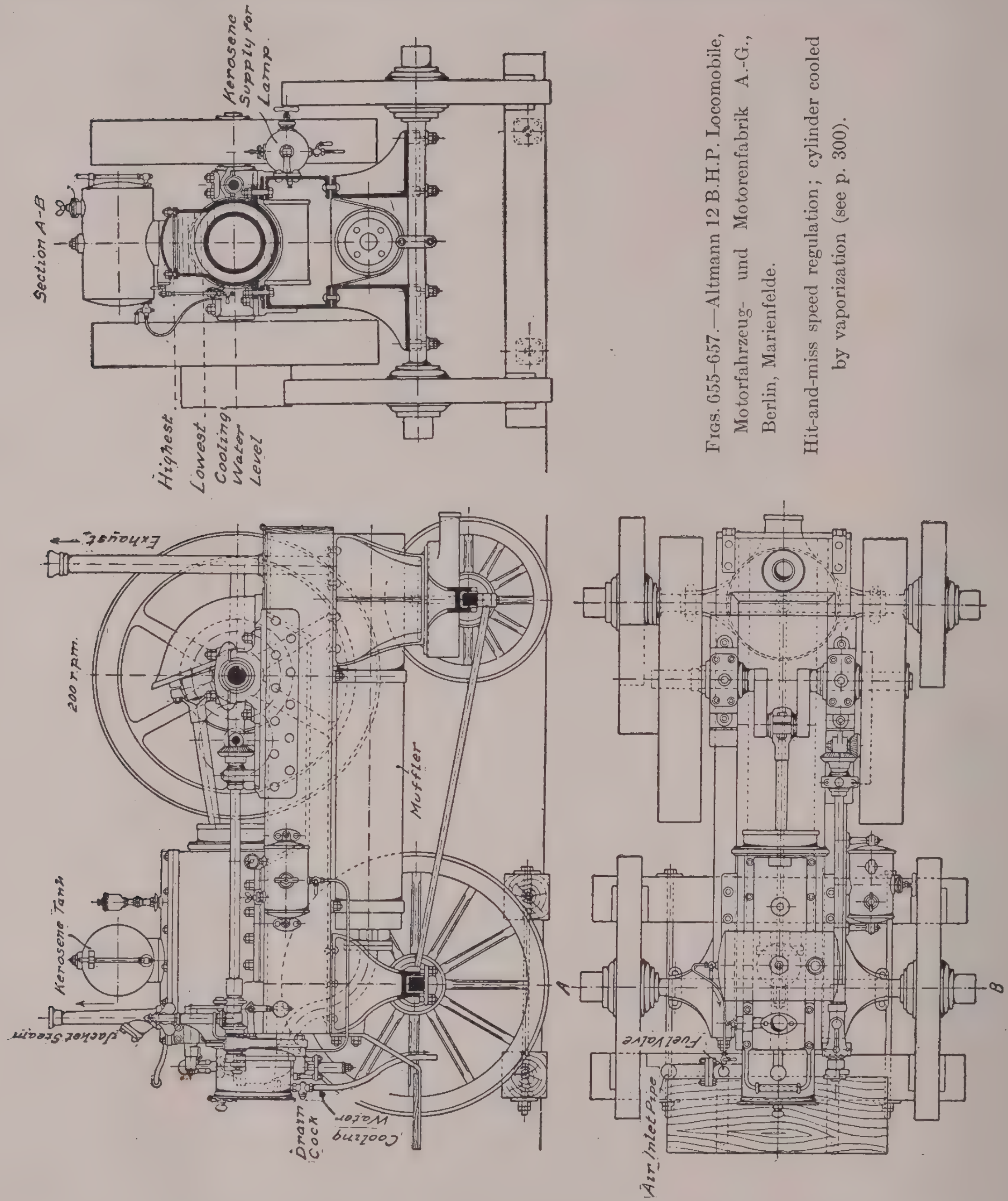


Fig. 654.—Wenzel Locomobile, built by Fr. Zimmermann & Co. Akt.-Ges., Halle S.

Water cooled by extended surface radiators.

Friction-transmission gear.



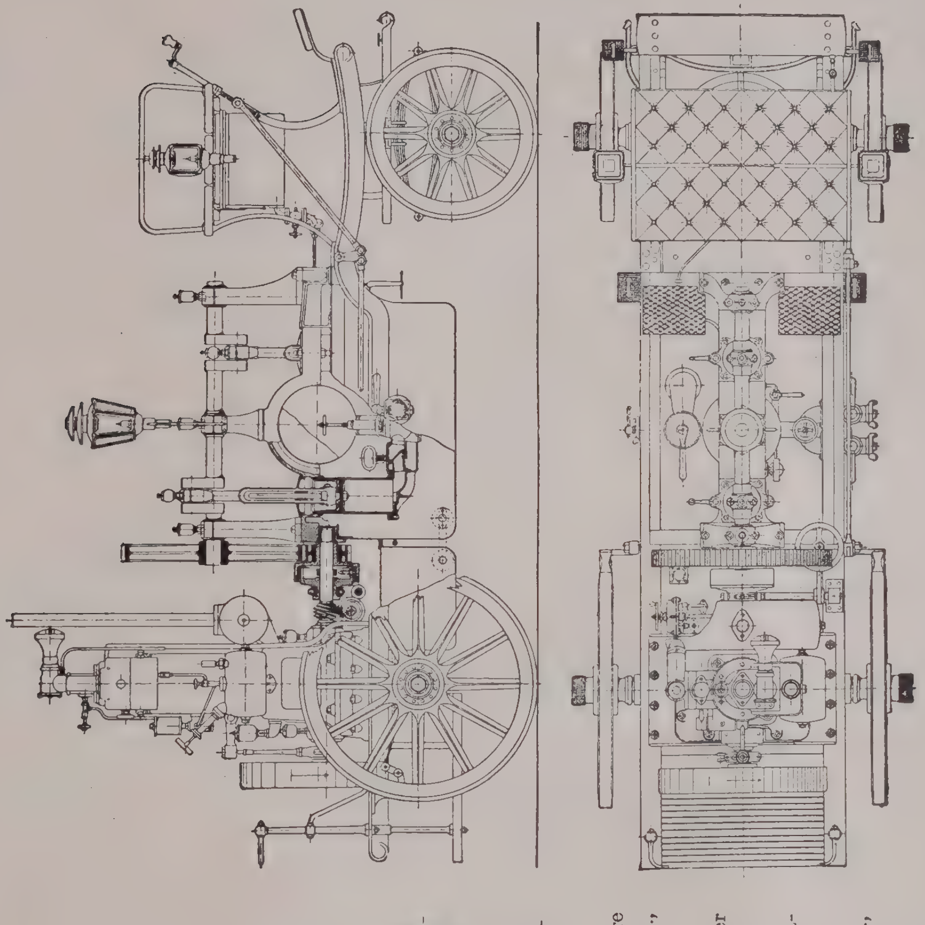
Figs. 655–657.—Altmann 12 B.H.P. Locomobile, Motorfahrzeug- und Motorenfabrik A.-G., Berlin, Marienfelde.

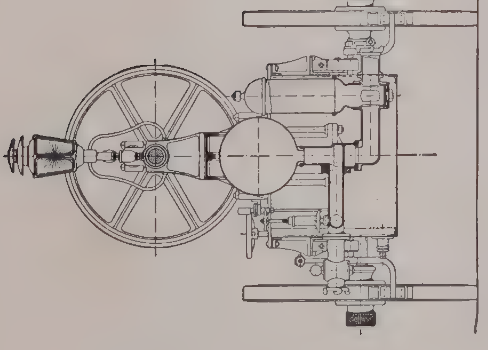
Hit-and-miss speed regulation; cylinder cooled by vaporization (see p. 300).

Table 108

DIMENSIONS AND WEIGHTS OF ALTMANN KEROSENE LOCOMOBILES

Rated B.H.P.	2	4–5	6–8	8–10	10–12	12–15	16–20	20–25
R.p.m	220	220	220	220	220	200	200	200
Approx. weight, lbs	2530	4400	5060	6600	7700	8800	4900	11000
Max. length (without tongue), ft.	7.22	10.65	10.65	10.65	10.65	11.50	11.50	11.50
Maximum height, ft.	3.60	5.90	5.90	5.90	5.90	5.90	5.90	5.90
Maximum width, ft.	7.22	7.22	7.22	7.22	7.22	8.20	8.20	8.20
Fly-wheel diam. and width in.	39.4×3	47.2×3.9	47.2×5.9	51.2×7.9	51.2×9.1	59 ×9.9	59 ×11	59 ×11.8
Belt pulley diam. and width in.	13.8×5.9	19.7×3.9	19.7×3.9	19.7×7.1	19.7×7.1	23.6×7.9	23.6×11	31.5×11.8





Figs. 658–660.—Gasoline Engine Fire Pump, Grether & Co., Freiburg, B., with 10 H.P. Deutz Engine.

Capacity of pump is 2.8 gallons per second, discharge pressure 90 lbs.

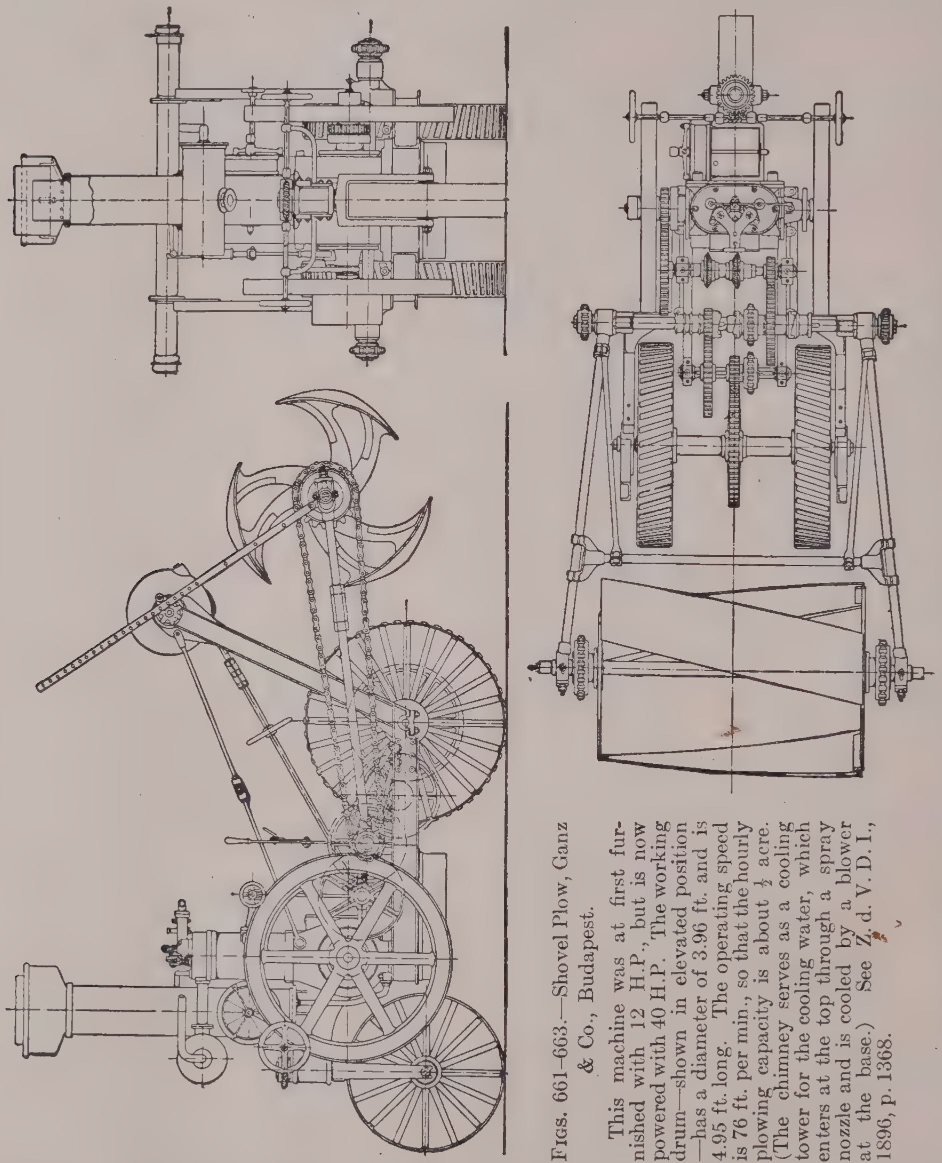
Dimensions of pump: 2 cylinders, dia. 5.9″, stroke 8.25″, r.p.m. 80.

(For further details see Z. d. V. D. I., 1896, p. 1314.)

Fire Pumps operated by gasoline engines are superior to steam fire pumps in that they are ready for immediate operation and, on account of their smaller weight, are easier to handle. To do away with the use of teams these pumps have also been constructed as automobile pumps.

II. Motor Plows

The oldest motor plow was built by Ganz & Co. and was introduced in 1896. This plow was direct-connected, that is, the implement itself was directly driven by the engine, and it represented the first useful solution of the long-standing problem.



Figs. 661–663.—Shovel Plow, Ganz & Co., Budapest.

This machine was at first furnished with 12 H.P., but is now powered with 40 H.P. The working drum—shown in elevated position—has a diameter of 3.96 ft. and is 4.95 ft. long. The operating speed is 76 ft. per min., so that the hourly plowing capacity is about ½ acre. (The chimney serves as a cooling tower for the cooling water, which enters at the top through a spray nozzle and is cooled by a blower at the base.) See Z. d. V. D. I., 1896, p. 1368.

Ganz & Co., following the patents of their former general manager, Mr. Mechwart, do not use the ordinary plow share, but a sort of "drum" plow, illustrated in Figs. 661–663. This is suspended at the rear of the locomobile and turns with moderate speed while the locomobile moves ahead.[1] A 12 H.P. gasoline engine used in the first types proved itself considerably too weak for rapid and sure operation in difficult soil. The power capacity was then increased step by step until the final motive power consists of a 40 H.P. four-cylinder vertical twin engine (each pair of cylinders in tandem) making 600 revolutions per minute.

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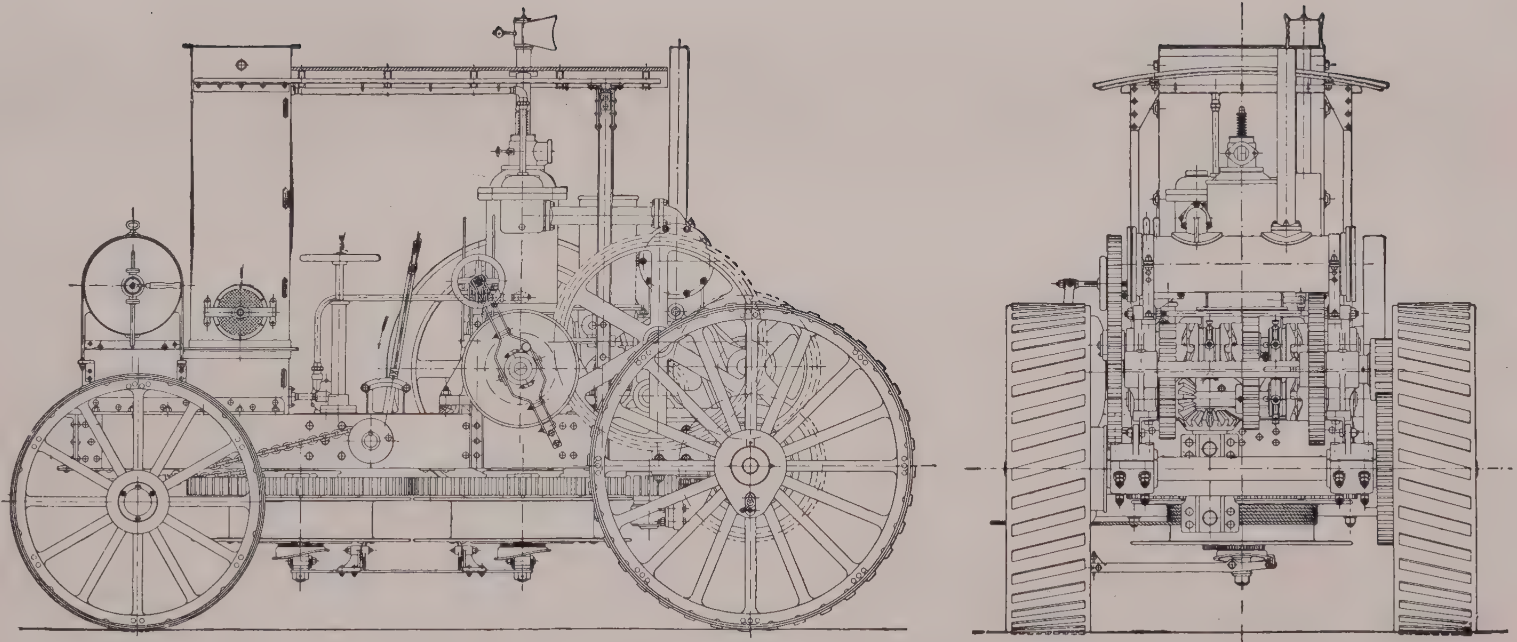
[1] For greater detail, see Z. d. V. D. I., 1896, p. 1367.

Of German gas-engine builders only the Motorenfabrik Oberursel builds motor-plows, Figs. 664 and 665. They pay special attention to the use of alcohol as fuel. The first attempts of this firm at building direct-acting motor plows do not seem to have been very successful, because the alcohol motor plow made by them to-day employs the so-called double machine system. This consists in placing a locomobile at each side of the field and having them draw a plow back and forth over the field between them by means of a wire rope. After each furrow the machines are moved sidewise the required distance.

The construction of such plowing machines does not offer any particular difficulties; the arrangement of the drives, etc., may be directly based on the designs of successful steam plows. The main point again is to make certain that there is ample engine capacity. Less than 20 B.H.P. should never be used, and it is better to commence with from 30 to 40. But under such conditions the motor plow no longer possesses any advantage over the steam plow regarding operating costs.

In spite of this fact agriculturists in general take a great deal of interest in the further development of the motor plow, especially with reference to the use of alcohol as a fuel. The principal thing expected from them is the general introduction of machine plowing which at present is not possible on account of the high cost of steam plows. The total capital cost of plowing machinery is of course least with the single machine system, for which reason, and in spite of the unfavorable results so far obtained with steam plows working on this system, attempts are constantly made to solve the problem with motor plows. It is true that the single machine system is best adapted to the motor plow because there is a pause of only about one minute, while the machine is changing to the next furrow, for every period of operation lasting from 3 to 5 minutes. The engine is therefore in use a large proportion of the time. In the usual two-machine system, on the other hand, each engine works for a period of from 3 to 5 minutes, plowing one furrow, and then rests for a period lasting from 5 to 8 minutes, while the opposite engine plows the next. Now it is well known that an oil engine is quite sensitive against such continued interruptions in the operation and starts again only with comparative difficulty. Hence it becomes almost necessary to keep the engine in motion also during the idle periods, which of course seriously affects the economy of the entire system. The single machine system is therefore not only cheaper as to first cost, but also more economical in operation.

As points in favor of the motor plow may be mentioned their lighter weight and the lower demand for fuel and water, as compared with steam plows. The fuel cost of moving them around from one place of operation to another is consequently less. Under certain circumstances these advantages may be more than sufficient to overbalance the greater costs of the kind of fuel used. Considerations of this kind of course by no means assure the future of the motor plow. The market for them is probably not extensive enough to lead manufacturers to undertake any extended and costly series of experiments, especially not at this time when the gas engine industry finds an apparently much more promising field in the building of blast-furnace gas engines, suction-gas plants, etc.



FIGS. 664, 665.—Alcohol Motor Plow, 20 H.P., Motorenfabrik Oberursel.

MAIN DIMENSIONS.

Front wheels: dia, 56″; tire, 18″.
Rear wheels: dia, 70″; tire, 18″.
Wheel base, 134″; wheel tread to center of tires, 85″.
Height to upper edge of platform, 47″.
Height of roof, 122″.
Length of frame, 153.5″; frame girders are channel iron, 12″×4″.
Dia. of rope drum, 31.5″; peripheral (plowing) speed 2.0 to 2.6 ft. per sec.
Cooling tower for jacket water: width, 37.5″; height, 67″; depth, 21.5″.

III. Motor Vehicles

This term includes automobiles and automobile trucks, for the transportation of passengers and freight, and motor boats. In a wider sense it also comprises self-propelled fire engines, traction engines, etc. The latter, however, will be treated, together with motor railways, under the next heading.

Concerning automobile engines proper, the author refers to his little special treatise on this type of engine.[1] In addition some of the later constructions are given below. The *power required* by an automobile may be estimated with sufficient accuracy by allowing 2.2 B.H.P. for every 1000 lbs. of total weight and for a speed of 10 miles per hour, assuming average roads and weather conditions.[2] It is not possible to give a generally applicable method of computing the power required on account of variation in the resistance encountered and the general uncertainty regarding them. Of the power developed in the engine, from 20 to 50% (and in especially bad cases even more) are lost in transmission between engine and driving wheels. The average powering of motor vehicles shows a steady increase. At present light automobiles are supplied with from 6–10 H.P., the heavier cars have from 15–20 H.P., while some of the racing cars show from 60–80 H.P.

All motors vehicles must be able to reverse their direction of motion (to back up). But since up to now no successful scheme has been worked out to reverse the engine itself, this can only be done by means of some auxiliary apparatus. Such change or reversing gears are interposed between the driving axle and the crank-shaft. In the case of *motor boats*, a device very often used is the reversible propeller in which, by varying the position of the blades, the direction of motion of the boat may not only be reversed, but intermediate speeds either ahead or astern may be obtained. But all of these reversible propellers when used under considerable stress any length of time develop bad faults, so that their use for large powers is almost out of the question. As soon as the power exceeds 60 or 80 H.P. there is no doubt that the ordinary screw with solid blades will give better service, and motion astern will then have to be obtained by the use of some reversing gear or clutch. This does not mean to say that the latter scheme is perfect, but it is the most reliable as long as a good way of reversing the engine itself is as yet not available. Until the latter problem is solved it is quite likely that the use of internal-combustion engines in marine practice will be largely confined to the smaller capacities.

In consideration of the fact that the crank-shaft must be direct-connected to the propeller shaft, the engine speed should not exceed from 250 to 350, as a maximum 400, turns per minute. Between engine and propeller shaft there should be a clutch in order to be able to start the engine when not under load. In the case of freight and tug boats the clutch should be so designed as to allow of somewhat continued slipping of one half upon the other, in order to bring the engine up to load gradually. A sudden taking hold of the clutch may bring the engine up standing. Where reversible propellers are used this of course may be done by gradually changing the position of the blades. The fly-wheels for these engines should combine maximum moment of inertia with smallest diameter. This has often led to making them solid disks with wide heavy rims. It is usual to deliver marine gas engines together with their auxiliaries mounted on structural steel frames to the builders of the hull, together with directions concerning the erection. The following extract from one of the older sets of "Directions for the Erection" of Capitaine-Grob marine engines may serve as an example of this practice.

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[1] Konstruction und Betriebsergebnisse von Fahrzeugmotoren für flüssige Brennstoffe, Berlin, 1901. Julius Springer.

[2] For the mathematical derivation of this statement see the author's treatise referred to, p. 44.

"The engine represents a certain weight whose position must have some influence on the position of the hull in the water. The latter should in all cases be the natural one, that is, not down by the stern and least of all by the head. The factors that must also be considered in the problem are the effect of the cargo (passengers or freight) and the fact that a boat under headway always exhibits a tendency to lower the stern. The proper consideration of all these things determines the right location for the engine, which may be at the middle, that is, coinciding with the displacement center of gravity, or may be ahead or astern of the same. Other qualities of the boat, such as speed, etc., are not materially affected by engine location.

In boats of ordinary proportions of hull, the engine is best located nearly in the middle. Care should of course be taken to see that the center line of the shaft comes directly over the keel to avoid unequal distribution of the load.

To securely place the engine it is necessary to connect a number of frames by means of a foundation plate so that both engine and thrust block may be placed on it, thus rigidly connecting the two.

It is of the greatest importance to have both engines and thrust block on a common foundation plate.

The engine should be set as far down as the fly-wheel permits, and its position should be inclined enough to have the center line of the shaft coincide with that of the stern tube.

The lower part of the fly-wheel should be encased water-tight in order to protect it against bilge water.

When no special engine room or cabin is provided, that is, when the boat is open, it is advisable to encase the engine. To leave it without any protection whatever might lead to serious rusting. Where the boat carries a cabin, it pays to partition off a portion of it to accommodate the engine. In the larger boats of course separate engine rooms are provided as in steam vessels.

The exhaust muffler is best placed ahead of the fly-wheels between two of the frames. The pipe from the engine to the muffler should of course be as short as possible. The pipe from the muffler may be laid along a longitudinal wall to the stern, having near the end a slight incline to avoid the water. To make the exit horizontal might cause the gas to give trouble, neither is it advisable to exhaust under water.

If in an open boat the motor is encased, ample opportunity should be given for free access to all of its parts by means of doors or covers.

The fuel tank is in most cases best placed in the bow, and should be made as large as possible. In order to utilize all the available space, the tank is usually made to conform to the shape of the vessel and is therefore best made on the spot. The engine is furnished cooling water by means of the pump attached. The suction pipe of this pump is directly connected to a sleeve passing through the bottom of the hull. The warm water is discharged in a similar way. The suction opening should be provided with a screen to prevent the access of foreign bodies to the pump. Both suction pipe to pump and outlet for the jacket must be furnished with a three-way cock, so that the water may be shut out when the engine is to be taken apart and also to be able to drain the pump and jackets in winter.

If the boat is to be used on salt water, the engine cylinder should be cooled by vaporization with fresh water, because the salt water is apt to deposit scale on the jacket walls and to corrode the metal.

The *power required* by motor boats under ordinary circumstances is given in Table 109. The power required by tug boats in water comparatively free from current (canals for instance) is estimated at .1 B.H.P. for every ton of towage and for speeds of from 3 to 5 miles per hour. The figures, however, apply only under favorable average conditions; stormy weather, counter-currents, bad design of hull, etc., cause considerable increase in the power demand.

TABLE 109
DIMENSIONS AND PERFORMANCES OF MOTOR BOATS

Engine B.H.P.	Length of Boat, Feet.	Speed, Miles per Hour.	Cargo.		Towing Capacity, Tons.	Draft, Ft.
			Freight, Tons.	Passengers.		
1	20.0	3.8	1.5	8	...	....
2	23.0	5.0	2.0	12	...	1.64
3	26.4	5.6	3.0	16	30	1.64
4	29.5	6.2	4.0	22	40	1.97
5	32.8	6.9	5.0	26	50	2.30
8	36.0	8.1	6.0	30	80	2.62
10	39.4	8.7	8.0	36	100	3.94
12	42.6	9.3	8.0	36	120	4.10

The *fuel consumption of automobile engines* varies from .16 to .24 lb. of gasoline per ton mile, for the average condition of road and depending upon size and quality of engine and car.

For the larger *boat engines*, fuel costs alone amount to about .04 cents per ton mile with suction-gas engines and .20 cents with oil engines. Opposed to this are a cost of from .24 to .32 cents per ton mile for towing by horse or mule team and of from .28 to .40 cents per ton mile for electric towing.[1]

The ratio of "steaming radius" as between steam vessels and motor boats is from 1:6 to 1:8 (1 cu.ft. of coal is estimated to furnish 8½ H.P. hours, the same volume of fuel oil supplies 58 or 60 H.P. hours). Suction-gas installations require more space for both power plant and fuel than oil engines, hence the economic superiority of the suction-gas plant is the deciding factor. The first gas tugs were built by the Gasmotorenfabrik Deutz; lately, however, Emil Capitaine has also taken up this particular field of usefulness for the gas engine with a great deal of energy.[2]

Designs of Automobile and Launch Engines:

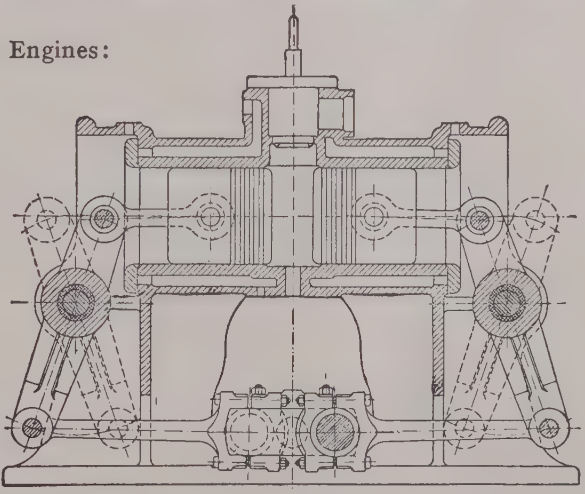

FIG. 666.—Opposed Engine, Wenzel Type, built by Fr. Zimmermann & Co., Halle S.

Does not differ materially from the engine shown in Fig. 667. Used mainly in kerosene locomobiles, rated capacity 8–12 H.P.

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[1] Note that these figures are quoted from German practice.

[2] See Capitaine's paper before the Schiffbautechnische Gesellschaft, 1904, printed in the year book of the Sch. G., 1905, Berlin, J. Springer.

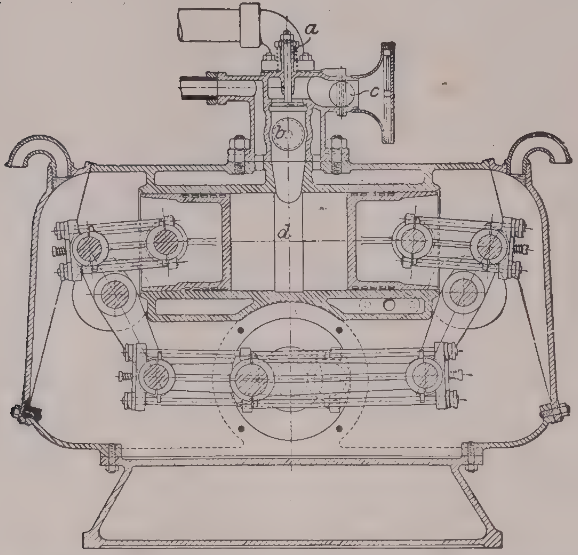

FIG. 667.—Opposed (Balanced) Engine, Capitaine Type, built by the Leipziger Motorenfabrik vorm. Swiderski.

(The combustion chamber, *d*, is contained between the heads of the opposed trunk pistons, which latter are connected to cranks 180° apart. Inlet valve *a* is automatic, while the exhaust valve *b* is operated by an eccentric through two-to-one gearing, *c* is a throttle valve for air. The vaporizer acts as igniter and is located opposite *b*). This construction results in good balance, so that high speed is permissible.* Capacity up to 12 H.P., mostly used as a launch engine.

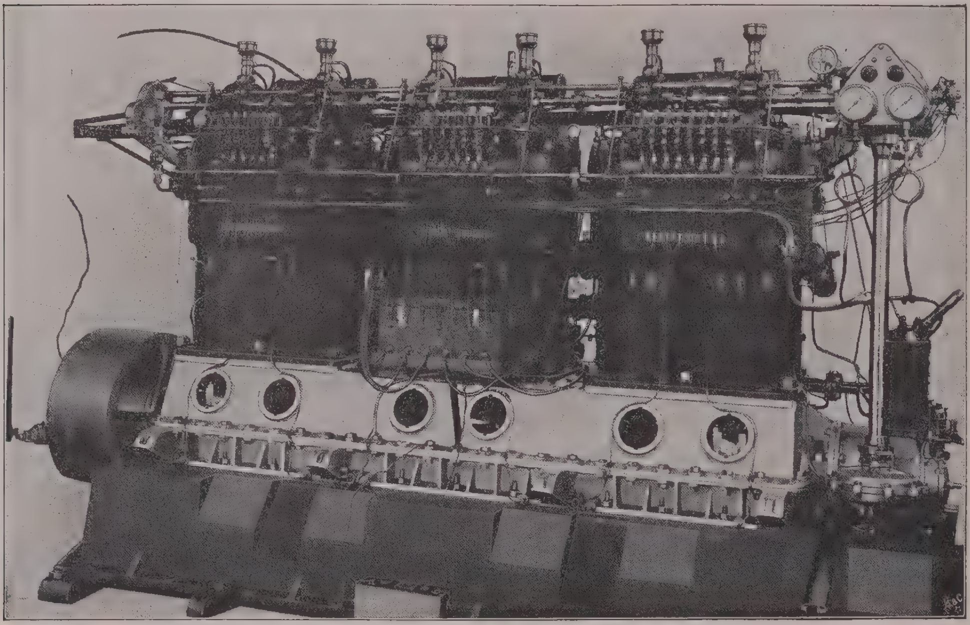

FIG. 668.—Six-cylinder Loutzki Boat Engine.

Rated H.P. = 500 at 900 r.p.m. ($D=12''$, $S=7.9''$). Fuel is alcohol or gasoline. Weight approximately 5280 lbs. when crank case is made of "Metorit" (aluminum alloy). Each cylinder has two igniters, one compressed air starting valve, and one combined inlet and outlet valve, as per construction shown in Fig. 669. The latter valve, on account of its small lift (= .5″ while the dia. = 11.0″) is supposed to admit of high rotative speeds. (Valve disk *a* is internally cooled and operated from shaft *a′*; concentric valve *b* is operated by the two rods *b′* and seats at *c*.

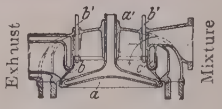

FIG. 669.—Combined Inlet and Outlet Valve, Loutzki.

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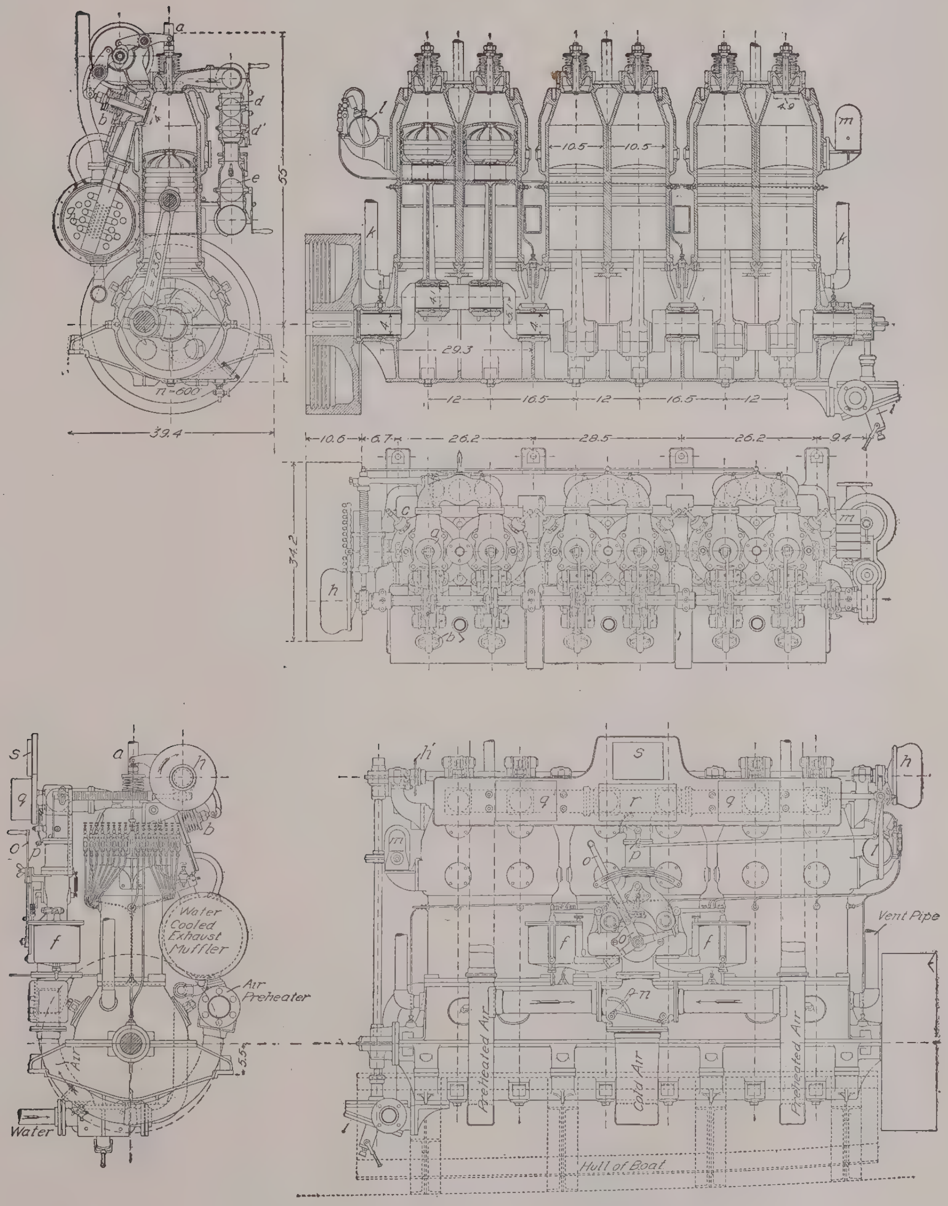
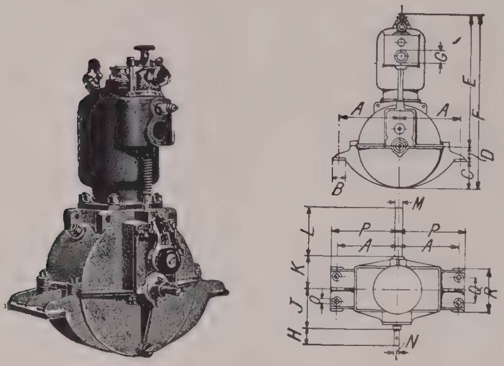
* The inventor of this construction, intended to secure balancing, is Ch. Brown, who applied it to steam engines (see German Patent No. 72625, 1893). Capitaine was the first to apply the principle to internal-combustion engines.



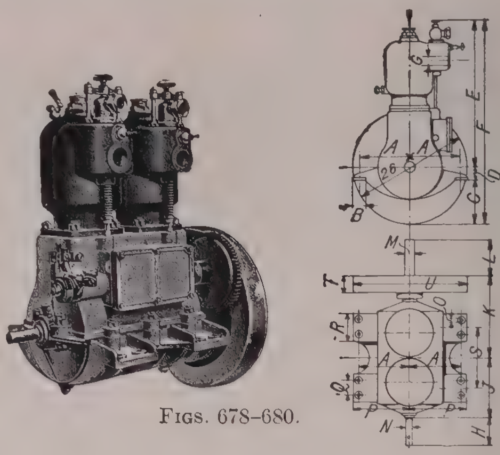
FIGS. 670–674.—Six Cylinder Loutzki Boat Engine.

Rated Capacity 300 B.H.P. at 600 r.p.m. Weight complete 8800 lbs. Fuel consumption at 320 B.H.P. (max. capacity) was found to be .573 lb. of gasoline or .793 lb. of alcohol per B.H.P. Hour.

SYMBOLS: *a*, inlet valve; *b*, exhaust valve; *c*, spark plug; *d*, *d'* and *e*, throttle valves for air, gasoline vapor and mixture; *f*, carburetors; *g*, exhaust muffler, water cooled externally, internal air pre-heater; *h*, shaft governor (controls throttle valve, *d'*); *h'*, commutator for ignition system; *i*, circulating pump; *k*, vent pipe: *l*, oil cups for the pistons; *m*, magneto; *n*, control lever to adjust supply of cold and pre-heated air; *o*, same for the carburetors; *p*, governor reach rods; *q*, spark coils; *r*, interruptor; *s*, switchboard.



Figs. 675–677.



Figs. 678–680.

Table 110

DIMENSIONS AND WEIGHTS OF WENZEL AUTOMOBILE AND LAUNCH ENGINES, BUILT BY H. KÄMPER, G. M. B. H., BERLIN, W.

Single-cylinder Engines, Figs. 675–677

(Dimensions in inches)

Model.	Rating B.H.P.	R.p.m.	Bore and Stroke, Inches.	W't. lbs.	*A*	*B*	*C*	*D*	*E*	*F*	*G*	*H*	*I*	*K*	*L*	*M*	*N*	*O*	*P*	*Q*	*R*
A	3	800–1000	3.54×3.94	143	8.5	1.56	3.9	1.77	18.7	24.4	1¼	1.97	5.5	4.37	7.9	.98	.71	.51	9.3	3.31	5.9
B	4	800–1000	3.94×3.94	165	8.3	1.56	3.9	1.77	19.1	24.8	1¼	1.97	5.9	4.77	6.3	1.10	.71	.50	9.0	3.54	5.9
C	6	800–1000	4.33×4.73	220	9.6	1.97	4.7	1.97	22.4	29.1	1½	1.97	6.3	5.42	6.8	1.26	.87	.51	10.6	4.18	6.7

Table 111

Two-cylinder Engines, Figs. 678–680

Model.	Rating B.H.P.	R.p.m.	W't. lbs.	*A*	*B*	*C*	*D*	*E*	*F*	*G*	*H*	*I*	*K*	*L*	*M*	*N*	*O*	*P*	*Q*	*R*	*S*	*T*	*U*
D	8	750–800	264	7.1	.78	6.1	1.77	23.2	29.3	1¼	3.9	8.1	12.0	3.9	1.42	.98	.51	7.9	1.97	3.9	8.3	2.36	15.7
E	10	750–800	319	8.4	.98	5.9	1.97	24.6	32.7	1½	4.3	8.9	13.4	4.1	1.65	1.18	.51	9.5	1.97	4.3	9.1	3.34	15.7
F	12	750–800	440	11.0	1.18	5.9	2.36	25.6	35.4	2	4.3	11.2	13.8	5.9	1.81	1.18	.63	12.2	2.76	4.7	9.8	3.9	19.7

IV. Motor Locomotives and Gas Railways

Motor locomotives (traction engines, etc.) nearly always use liquid fuel. This branch of the gas engine field has received but little attention in Germany and on the continent in general, although many gas traction engines are in use in English and other colonies. It is quite likely, however, that the attempts which have for several years back been made by a number of the world powers to adapt the traction engine to army service will do much to stimulate the further development of this branch of the industry. The prize competition instituted in 1901 by the Prussian War Office, and from which a great deal was expected, had practically no results, mainly because the requirements imposed upon the competitors were impossible of fulfilment at that time. It served the purpose, however, of giving an impetus to the entire industry, which will very likely result in some benefit to the development not only of the traction engine but of the entire field of self-propelled vehicles as well.

Gasoline or oil locomotives (running on rails), the building of which was to the author's knowledge first taken up by G. Daimler some twenty or more years ago, have at the present writing reached a high state of development and are in extended use on industrial railways (narrow gauge) of all kinds. Their average power capacity does not at present exceed from 12 to 15 H.P., except in rare cases. But a good solution of the problem of direct reversibility would also do a great deal toward further development in this case.

Gas locomotives burn illuminating gas compressed to from 120 to 225 lbs. per sq.in. This gas is carried in tanks, which hold sufficient fuel for certain limited distances, the pressure being reduced before use to nearly atmosphere by means of automatic regulators.

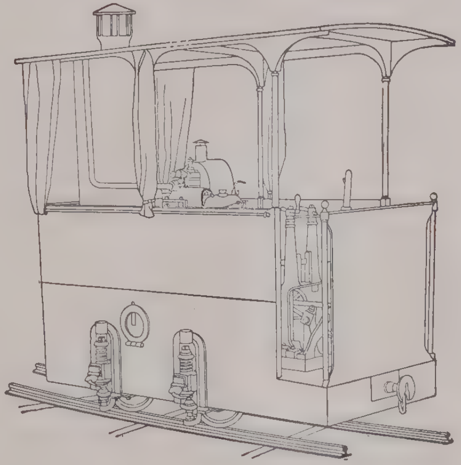


Fig. 681.—Main Dimensions and Capacities of Motor Locomotives, Fig. 681, built by Capitaine-Grob Co. Normal speed 5 miles per hour.

The weight in service depends upon the rail gauge. A change in gauge changes the weight.

The total loads moved (exclusive of weight of engine), as given below, are based upon the assumption that the cars as well as the track are in fair condition, and that the weather conditions are normal.

TABLE 112

Nominal horse-power		4	6	8	10
Number of driving axles		2	2	2	2
Diameter of wheels, ft.		1.97	1.97	2.30	2.30
Wheel base, inches		33.5	33.5	35.4	37.5
Minimum gauge, inches		17.7	19.7	19.7	23.6
Supply of cooling water, pounds		110	110	165	220
Supply of kerosene (sufficient for ten hours), gallons		5.2	6.5	9.1	10.4
Weight of locomotive in service, approx., pounds		3520	4400	5500	7270
Effective tractive effort, pounds		275	374	517	725
Total loads moved in tons (exclusive of weight of locomotive) on grades of	1:20 =50%	.8	1.0	1.5	2.0
	1:25 =40	1.0	1.5	2.0	3.0
	1:30 =33.3	1.5	2.0	3.0	4.0
	1:40 =25	2.0	3.0	4.5	6.0
	1:50 =20	3.0	4.5	7.0	8.0
	1:60 =16.5	4.5	6.0	8.0	10.0
	1:80 =12.5	6.0	7.5	10.0	13.0
	1:100=10	7.0	9.0	13.0	16.0
	1:200=5	11.0	14.5	20.0	25.0
	1:500=2	15.0	21.0	29.0	35.0
	1:∝ =0	20.0	28.0	39.0	50.0

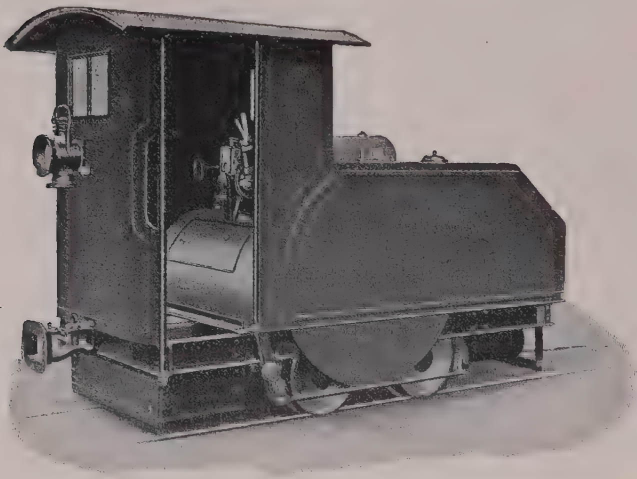


Fig. 682.—Deutz Motor Locomotive.

The smaller sizes of Deutz gasoline mine locomotives differ somewhat regarding the construction of the transmission gear, etc., from the types shown in Figs. 682–684. The external dimensions are such that the completely erected locomotive can pass through the mine shaft. (See Z. d. V. D. I., 1902, p. 490).

Table 113

PRINCIPAL DIMENSIONS AND CAPACITIES OF DEUTZ LOCOMOTIVES

Type	Mine or Agricultural Railway.	Mine or Agricultural Railway.	Mine or Agricultural Railway.	Street Railway or Switch Engine.	Mine or Agricultural Railway.	Street Railway or Switch Engine.	Mine or Agricultural Railway.	Street Railway or Switch Engine.	Street Railway or Switch Engine.
Size of engine, B.H.P.	6	8	12		16		24		32
Tractive effort on level track, based on a speed of m. per hr...	2.5–4.5	2.5–5.6	2.5–7.5	2.5–7.5	2.5–7.5	2.5–7.5	2.5–7.5	2.5–7.5	2.5–7.5
=appr. lbs.	572–330	770–352	1152–384	1100–352	1560–505	1540–462	2420–770	2420–770	3180–1010
Length incl. buffer, ft..	9.05	9.80	11.65	10.5–12.30	11.80	....	13.10	11.80–13.10	13.10–14.45
Minimum width, ft. . .	2.63	2.86	3.35	6.65	3.78	....	4.43	7.23	7.23
Height without roof, ft.	5.02	5.08	5.42	8.85 *	5.91	....	6.43	8.85 *	8.85 *
Minimum gauge, in. . .	17.8	18.7	19.7	29.5	23.6	29.5	23.6	29.5	29.5
Weight in service, single transmission, lbs.	5940	6800	9700	....	....	....	....	....	....
Do., double transmission, lbs	....	8150	10 500	14 200	12 300	16 500	15 400	19 800	25 300

* With roof.

The first gas street railway was put in operation in Dessau in 1894. The cars used were furnished with a two-cylinder opposed (Luhrig) engine which was placed under one of the seats lengthwise of the car so that the fly-wheel came about the middle and outside of the car frame.[1] The power originally used was 8 H.P. which was afterward increased to 12 H.P. This was transmitted to the driving axles at various speeds by means of a compound gear transmission. The gas tank was located under the opposite seat, while the cooling water tank was placed on the roof. The gas supply carried amounted to about 28 cu.ft. under a pressure of about 120 lbs., when this was nearly used up the cars had to stop at special compressing stations for a new charge. The motive power on this road was changed to electricity a few years ago.

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[1] Drawings and description in Z. d. V. D. I., 1895, p. 1009.

Operating Results. (*a*) The gas consumption on the Dessau Street Railway amounted to from 11.5 to 17.7 cu.ft. of gas per car mile, varying with the grades on the particular run. The average speed was 6.25 miles per hour including stops.

(*b*) According to data furnished by Kemper,[1] the cost of building gas railways is about $30,000 per mile, and the total operating costs about 6.5 cents per car mile (assuming the price of gas at 85 cents per 1000 cu.ft.) Compared with this, the same authority states the cost of building an electric road to be about $36,000 per mile, and places the total operating costs at 7.8 cents per car mile. In spite of this apparently superior showing, gas railways have found very little application. The reasons for this probably have nothing to do with the economic side of the case, but are more likely found in certain external difficulties and inconveniences experienced in operation. The gas engine makes larger demands as far as space is concerned than the electric motor, and further, although it may be comparatively slow running, a reciprocating engine causes vibrations still distinctly felt in spite of nearly perfect balancing. The heat radiated by it, as well as gas and lubricating oil vapors, are apt to cause annoyance in the interior of the car, while the exhaust gas and oil thrown around may do the same thing on the outside. This was particularly noticed in the case of the Dessau railway by persons who had been used to electric cars.

(*c*) A self-propelled car often found on small private railways is the Daimler gasoline passenger car. This car, being independent of any charging station, is therefore more simple to operate and shows better results than the car operated with gas. This is quite clearly shown by the following table, which is taken from a report on Daimler cars made by the Kirchheimer Eisenbahngesellschaft:

TABLE 114

GASOLINE COSTS, DAIMLER NARROW-GAUGE GASOLINE CAR

Weight of car with 5 to 6 B.H.P. engine	7050 lbs.	Trip out: Time 37 min. corresponds to a speed of approximately	9.4 miles per hr.
Wheel base	4.60 ft.	Gasoline used: In vaporizer	.416 gallon
Contains seats for	10	In ignition lamp	.052 "
and standing room for	10	Total	.468 gallon
Trial trip between Esslingen and Plochingen, with a load of 7 passengers and 1340 lbs. of iron, equivalent to 15 passengers.		Hence cost per mile $=\frac{.468}{5.62}\times 20=1.62$ cents.	
Length of line	5.62 m., of which 4.37 miles are level, while the remaining 1.25 miles show grades of $\frac{1}{50}$ to $\frac{1}{200}$.	Return trip: Time 34 min., equivalent to a speed of, approximately	10 miles per hr.
		Gasoline used: In vaporizer	.350 gallon
		In ignition lamp	.052 "
		Total	.402 "
		Hence cost per mile $=\frac{.402}{5.62}\times 20=1.43$ cents.	

(*d*) The computation of operating costs presented by the following table is given by the Gasmotorenfabrik Deutz for the type of locomotives illustrated by Figs. 683 and 684. They are based upon the actual expenses connected with the operation of two mine locomotives of 6 and 8 H.P. respectively, and of two farm railway locomotives of 8 and 12 H.P.

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[1] Journal für Gas Beleuchtung, 1893, p. 505. Note that these and the following figures apply to German practice.

TABLE 115

OPERATING COST OF DEUTZ GAS-ENGINE LOCOMOTIVES

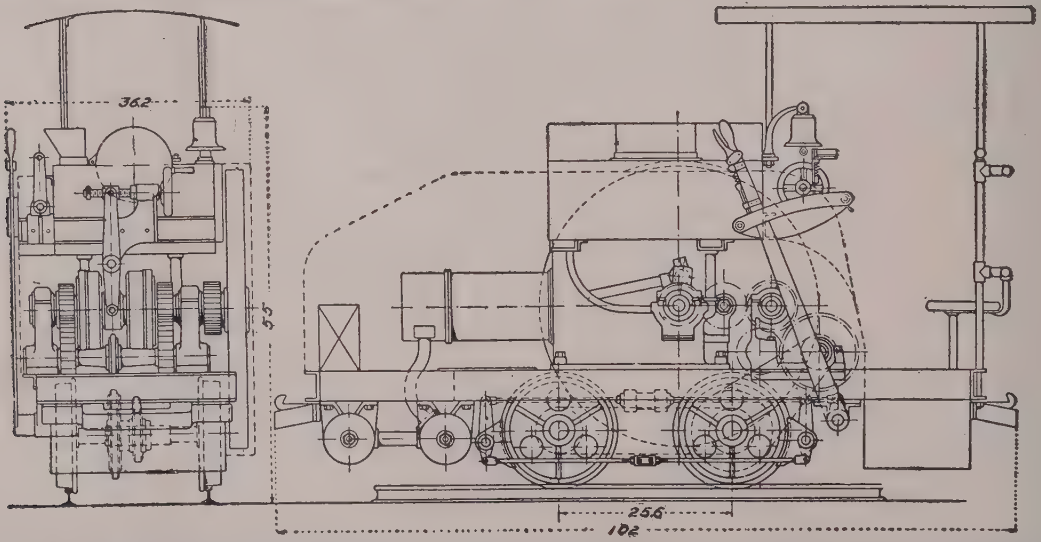
Owner.	Gräflich v. Arminsches Forstamt.	Vereinigte Königs- und Laurahütte, Akt. Ges. Laurahütte.	Verein. Gesellschaft für Steinkohlenban in Wurmrevir, Kohlscheid.	Herzogl. Württembergisches Rentamt Karlsruhe, O.-S.
Size	6 H.P.	8 H.P.	8 H.P.	12 H.P.
How used	Mine	Mine	Farm railway	Brickyard railway
Length of haul, miles	.375	.28	.485	1.87
Grade	Level	Level	1:62.5	1:33
Load, tons	250 } 10 hr. shift	400 } 9½ hr. shift	225 } 10 hr. shift	150 } 11 hr. shift
Capacity, ton miles	93.8	112	109	280
First cost:	Type CI.	Type CI.	Type CII.	Type CII.
Locomotive, dollars	1675	1875	2135	2490
Charging stations, dollars	75	75	112.5	512
	1750	1950	2247.5	3002
Yearly costs: (300 days)				
Interest, depreciation, repair, 17%, dollars	298	331	382	510
Engineer, dollars	300*	300	300	300
Fuel, dollars	135	202	190	256
Oil and waste, dollars	42.5	46	45	53.5
	775.5	879	917	1119.5
Total operating costs per ton mile, cents	2.77	2.62	2.81†	1.33

* Very low for American conditions.

† The comparatively high cost is directly due to small demand on the locomotive.

Compared with the above table, the cost of doing the same work with horses would have been from 4 to 8 cents per ton mile, depending upon locality and hauling distance.

Designs of Gasoline Locomotives:



FIGS. 683, 684.—Gasoline Mine Locomotive, Gasmotoren-Fabrik Deutz. Gauge 21.6″, Engine Capacity 8 H.P., Speed 2.8 and 5.6 miles per hour. Corresponding Tractive Efforts 570 and 308 lbs. respectively, on level track. (Engine and transmission gear are enclosed in sheet steel hood indicated by dotted line.)

PART IV

THE GAS ENGINE FUELS AND COMBUSTION IN GAS ENGINES

A. FUELS

In the operation of internal-combustion engines we consider only the fuels that will form ready mixtures with air and which will burn without any considerable residue. These requirements are most nearly met by the gas fuels, and for that reason these have been almost exclusively used in the operation of engines even from the first stages of the development.

Natural gas, which is obtained from wells in various localities of the world, need not be considered here.[1] Most of our *fuel gases* are obtained either through distillation or through gasification of solid fuels. The *fuel oils* are converted into vapor or spray before being burned in engines. Between the original condition of the fuel and the state in which it is finally used there is consequently an act of transformation which may be brought about in apparatus either entirely independent of the engine installation, in apparatus clearly auxiliary to the engine, or even in the engine itself. Examples of these three methods of transformation are found in the manufacture of illuminating gas, the making of producer gas, and the production of kerosene vapor. If the transformation takes place outside of the engine, the process usually involves, through loss of fuel and heat and through operating cost, an increase in the *specific cost of heat* of the fuel consumed. Any fuel consequently possesses its lowest heat cost in its original shape, and other things being the same, would therefore give the highest economy when used directly in this condition. In the operation of gas engines, however, this is possible only with liquid fuels.

The choice of solid fuels suitable for the production of power gas is, generally speaking, unlimited. The necessity, however, of making gas producers as simple as possible regarding construction and attendance, so that they may form a comparatively cheap auxiliary to the gas engine and may be taken care of by the same attendants, at once limits the choice of solid fuel for operation in the ordinary installation. Where these restrictions do not exist, that is where the producer installation may be built without reference to the cost of construction or of maintenance, it may be said that to-day every solid fuel is available for power. On the other hand—and that is the usual case—only such solid fuels can be considered whose properties with regard to composition, physical size, behavior in the producer, etc., are such as to allow of gasification in the most economic way and with the simplest means.

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[1] Translator's Note. This gas is yet largely used in American practice, and hence some constants are given on page 526.

In this respect the fuels consisting most nearly of pure carbon, that is, anthracite and coke, maintain their position as the most common fuels used in producer practice. All of the bituminous coals, when used in producers, as yet cause serious difficulties, either through high percentage of ash, presence of sulphur and tar-forming hydrocarbons, tendency to coke or clinker, irregular size, etc. However, the building of serviceable producers to handle the so-called brown coals or lignites, especially in the shape of briquettes, has been quite successful (see p. 282).

The following tables show the composition of a number of different fuels together with their heating values. The first table, No. 116, has been compiled entirely from

TABLE 117

Kind of Fuel.	Name or Location.	Proximate Analysis.				Heating Value per lb.	
		Moisture, %	Vol. Matter, %	Fixed C, %	Ash. %	of Combustible	of Dry Coal
Pa. anthracite	Treverton	.84	6.67	85.66	6.83	15195	14149
"	Lykens Valley (buckwheat)	.....	6.80	80.20	13.00	13680	11901
"	Jermyn (stove)	.....	6.08	82.90	11.02	13850	12324
"	Lackawanna	.....	5.00	84.00	11.00	13900	12371
"	Avondale	.....	6.00	87.78	6.19	14200	13317
"	Manville Shaft, Scranton	.....	6.12	86.50	7.38	14100	13059
Pa. bituminous	Antium	.....	18.54	70.16	11.30	15400	13660
"	Beaver Creek	1.50	34.33	55.42	8.75	14762	13450
"	Big Muddy	.....	83.20	58.20	8.60	14700	13436
"	Carnegie	1.45	36.42	56.20	5.93	14947	14047
"	Creedmoor	1.09	38.91	51.14	8.86	14983	13641
"	Pittsburg	.....	35.50	54.60	9.90	14200	12794
"	Reynoldsville	.....	24.67	69.96	5.37	16100	15235
"	Youghiogheny	1.40	33.13	60.82	4.65	16031	15275
W. Va. bit.	Clover Hill	1.34	31.70	56.83	10.13	14265	12800
"	Pocahontas	.80	18.30	73.65	7.25	15682	14536
Ohio bitumin's	Cambridge	2.43	37.79	50.36	9.42	14474	13076
"	East Palestine	.82	34.98	52.65	11.89	14603	12902
"	Hocking Valley	6.65	34.14	49.54	9.67	13972	12524
"	Mahoning	3.15	35.00	50.95	10.90	14578	12416
"	Waterford	1.55	37.29	53.34	7.82	14814	13637
Ind. bitumin's	Brazil	8.98	34.49	50.30	6.28	14542	13546
"	Lancaster	12.66	37.44	47.22	2.68	14251	13813
"	New Pittsburg	5.89	42.23	40.40	11.48	13973	12269
Illinois bitum's	Bryant	2.42	32.91	42.64	22.03	13157	10187
"	Big Muddy	7.40	28.30	53.90	10.50	13757	12212
"	Gillespie	12.60	30.60	45.30	11.50	12510	11947
"	Ledd	8.49	33.53	44.12	13.76	13876	11769
"	Mt. Olive	8.00	36.40	44.50	11.10	13158	11570
"	Peru	9.00	37.20	47.20	6.6	13603	12616
"	Streator	12.10	35.30	48.80	3.90	13580	12993
"	Vulcan	10.30	27.90	50.00	12.80	13817	11999
"	Wilmington (screenings)	.....	32.60	39.70	27.70	13200	9544
Lignite	Erie, Colorado	18.57	32.71	45.98	2.74	11360	10978
"	Cañon City, Colorado	6.56	36.74	47.93	8.76	11170	10122
"	Marshall, Colorado	13.19	37.84	46.43	2.54	11478	11143
Oven coke	Connellsville, Pa.	0.03	0.46	89.58	9.11	14211	12801
"	Fairmont, Pa.	0.30	0.62	85.78	11.46	14464	12534
"	Irivin, Pa.	.....	1.38	88.24	9.41	14447	12947
"	Kittanning, Pa.	.....	.....	87.22	11.43	14483	12632
"	New River, W. Va.	.....	.....	92.38	7.21	14427	13325
"	Pocahontas, W. Va.	0.29	0.49	92.58	6.05	14400	13441
"	Birmingham, Ala.	.....	.....	87.29	10.54	14290	12474

"Professional Paper No. 48, Report on Operation of the Coal Testing Plant of the U. S. Geological Survey at St. Louis, 1904."

Table 117 contains some additional information on Pennsylvania, Ohio, Indiana, and Illinois coals, together with a few figures on coke, lignites, and peat. This table has been compiled from Poole, "The Calorific Power of Fuels." The reader is referred to this book, which contains extensive tables of the properties of all kinds of fuel.

In the coking process some of the carbon in the original coal is always lost, as a part of it is used to form the volatile gases in conjunction with hydrogen (tar and hydrocarbons in the illuminating gas). The distillation commences at from 900 to 1450° F., and is therefore probably completed before the coal has reached the gasification zone of the producer, at least with small sized fuel. The main constituent, therefore, that is concerned in the combustion part of the gasification process is the carbon in the resulting coke, and this is often, as for instance in the case of lignite and peat, but a small percentage of the original fuel.

Consequently in such cases the yield of gas per pound of fuel is correspondingly low. High moisture content in the two fuels mentioned also sets a limit to their successful use in generators, see p. 269, and Section 5 in the Appendix.

Where fuel must be stored in large quantities or transported over considerable, distances, in certain cases by the motor itself, the question of *heat density*, that is the heat units contained in unit volume of the fuel, assumes importance.

Heat density is, roughly, inversely proportional to relative volume occupied, from which it follows that in this respect the liquid fuels stand first and the gas fuels last. It is of course possible to increase the heat density of gases by compression, but that by no means establishes anything like equality between them and the fuel oils. According to Table 118, for instance, rich oil gas would have to be compressed to 14 250 lbs. per sq.in., and illuminating gas even to 25 000 lbs. per sq.in., to make them volume for volume the equal of kerosene. From this it follows that where fuels must be stored or transported the liquid fuel is the ideal.

TABLE 118

WEIGHT, HEAT DENSITY, AND HEAT COST OF VARIOUS FUELS

	Gas Fuels.				Liquid Fuels.				Solid Fuels.			
	Illuminating Gas.	Oil Gas.	Producer Gas.	Blast-furnace Gas.	Kerosene.	Crude Oil.	Gasoline.	Alcohol, 90% by Volume.	Anthracite.	Brown Coal, or Lignite.	Gas Coke.	Cord Wood.
Wt. of 1 cu.ft., lbs. .	.032	.056	.068	.077	49.8	51.7	43.1	52.0	49.5	43.5	27.8	19.8
Vol. of 1 lb., cu.ft. . .	31.2	17.9	14.7	13.0	.0201	.0194	.0232	.0193	.0202	.0230	.0360	.0507
Heating value, B.T.U. per lb. . . .	17450	17800	1940	1370	18900	18000	19700	10000	11700	5400	12600	5000
B.T.U. per cu.ft. . . .	560	995	132	105	942000	928000	850000	519000	580000	235000	350000	99000
Vol. per 1000 B.T.U., cu.ft.	1.79	1.005	7.60	9.54	.00106	.00108	.00118	.00193	.00173	.00426	.00286	.0101
Heat density (kerosene = 100)	.06	.11	.014	0.11	100	98	90	55	62	25	37.5	11
Cost of 1 lb., cents . .	3.12	2.68	. . . .	. . . .	1.97	.73	2.62	5.80	.25	.10	.20	.38
Cost of 1000 B.T.U., cents	.179	.151	. . . .	. . . .	.104	.041	.133	.580	.0214	.0185	.0158	.0760
B.T.U. for 1 cent . . .	5600	6600	. . . .	. . . .	9600	24400	7500	1730	46800	54000	63200	13200

The above table is based on the following schedule of costs: Illuminating gas, \$1.00 per 1000 cu.ft.; oil gas, \$1.50 per 1000 cu.ft.; kerosene, 13 cents per gallon; alcohol, 40 cents per gallon; anthracite, \$5.00 per ton; lignite, \$2.00 per ton; coke, \$4.00 per ton; wood, \$3.00 per cord of 40 cu.ft. (8 ft. × 4 ft. × 15 in.).

I. Fuel Gases

For description of gas producer installations, see p. 268; for theoretical discussions of the gasification process, see Appendix, Section 5.

In regard to gas producing apparatus, gas-engine builders confine their attention to the gasification of solid fuels; where illuminating gas is used in an engine, it is nearly always made in *independent* plants by the usual distillation process. The reason for this is that the various kinds of producer gas are very easily made and cleaned with the simplest of apparatus. The manufacture of oil gas requires apparatus somewhat less complicated or costly than that used for illuminating gas, but the former gas, as far as its specific cost of heat is concerned, can only compete with the latter when the raw materials from which it is made are available at very low prices.[1] On this account oil gas, in spite of its great heating value and purity, generally finds application in engines only in places where it is used at the same time for some other and more important purpose.

1. Illuminating Gas. Illuminating gas is produced by the dry distillation of bituminous coals in closed retorts, which are heated to bright redness. Under these conditions the volatile parts of the coal are driven off, while the by-products are coke, ammonia, and a small quantity of graphite. Each 100 lbs. of average coal will yield from 400 to 480 cu.ft. of cooled gas, from 50 to 70 lbs. of coke, 4.75 lbs. of tar and from 8 to 10 lbs. of ammonia liquor having a specific gravity varying from 1.5 to 2.0° B. The heating of the retorts in an average plant requires from 15 to 20 lbs. of coke for every 100 lbs. of coal distilled. Allowing for the heating value of the coke so used, we find that of the original amount of heat contained in the coal, 20% is found in the gas and from 50 to 60% is recovered in coke and tar. The manufacturing cost of the gas of course varies with the size, arrangement and operation of the plant and with the cost of coal. The sale price usually ranges from \$1.00 to \$1.50 per 1000 cu.ft.

The *composition* of the gas is subject to constant change, even in the same plant. Its main constituents are hydrogen (45–48% by vol.), methane (35–38%), carbon monoxide (5–8%), and several per cent. of heavy hydrocarbons, carbon dioxide, and nitrogen. A number of gas analyses are given in Table 119. The contents of so-called illuminants in the gas has considerable influence on the heating value of the gas, but illuminating power and heating value do not bear to each other a sufficiently definite ratio to enable the determination of the latter on the basis of the former, as used to be done. The *heating* value varies from 450 to 680 B.T.U. per cu.ft., and in some isolated instances may reach 780 B.T.U. Since the introduction of Welsbach mantles and similar devices and owing partly also to the increasing cost of coal, the average heating value of the illuminating gas has shown a considerable decrease in the last few years. At present the mean value is probably not far from 560 B.T.U. per cu.ft. The best means of determining it is the calorimeter, but where the composition of the gas is very definitely known, it may be found with almost equal accuracy by computation.

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[1] This does not refer to the Lowe oil gas.

The density of illuminating gas varies from .35 to .45, (air=1.0), the specific weight is on the average .032 lb. per cu.ft. Gas holders are usually kept under a pressure of from 4 to 8 ins. of water. From these the gas enters the mains through the station pressure regulator under a pressure varying from 2 to 3 ins. of water, while the pressure in the most distant branch lines is in most cases only .8 to 1.2 in. For that reason it is usual to test out illuminating gas engines in the factory with a maximum gas pressure not to exceed .8 in. of water. The temperature of the gas in the mains is nearly always several degrees lower than the temperature of the outside air in summer, while in winter it may under certain circumstances be considerably higher than the air temperature. In mains put down rather deep the gas temperature is never found outside of the limits of 40 to 60° F., except on very cold or very hot days.

TABLE 119

ANALYSES OF ILLUMINATING GASES

	City.	Methane. CH_4.	Hydrogen. H.	Carbon Monoxide. CO.	Carbon Dioxide. CO_2.	Oxygen. O.	Heavy Hydro-carbons, C_nH_{2n}.	Nitrogen, (Remainder). N.
1	Berlin (coal from Upper Silesia)	32.7	49.7	9.5	2.5	....	4.6	1.0
2	Königsberg (Blochmann)	36.5	49.0	5.6	1.1	....	6.8	1.0
3	Magdeburg (Gasjourn. 1900, 87)	30.1	54.9	7.7	1.4	0.2	3.3	2.4
4	Dresden (Schöttler)	33.4	48.7	8.0	1.5	1.4	3.0	4.0
5	Hannover (Schöttler)	37.5	46.3	11.2	0.8	....	3.2	1.0
6	Frankfurt a. M. (Leybold)	32.6	49.8	8.8	2.3	....	4.0	2.5
7	Bonn (Clerk)	43.1	39.8	4.7	3.0	....	4.7	4.7
8	Heidelberg (Bunsen)	34.0	46.2	8.9	3.0	0.6	5.1	2.2
9	Aachen (v. Ihering)	34.2	54.0	5.2	1.1	....	3.3	2.2
10	Paris (Peclet)	33.1	50.1	6.3	1.5	0.5	5.8	2.7
11	London (Gaslight & Coke Co.)	37.6	48.0	3.7	....	0.3	4.4	6.0
12	Manchester (Clerk)	34.9	45.6	6.6	3.7	....	6.5	2.7
	Average	34.99	48.51	7.183	1.825	0.250	4.560	2.700

The heating value of No. 1 was approximately 560 B.T.U. per cu.ft.; of No. 3, 600 B.T.U. per cu.ft., and of No. 9, 628 B.T.U. per cu.ft.

TABLE 120

CONSTANTS FOR ILLUMINATING GAS

Composition of Illuminating Gas.		1 Cubic Foot of Gas Contains		H_u in B.T.U.	Air Theoretically Required.	
		Cubic Feet.	Pounds.		Cubic Feet.	Pounds.
Hydrogen	H	.4850	.00271	144.0	1.145	.0924
Methane	CH_4	.3500	.01562	333.2	3.285	.2651
Carbon monoxide	CO	.0700	.00546	23.9	.165	.0133
Heavy hydrocarbons	C_2H_4	.0450	.00351	70.4	0.639	.0516
Carbon dioxide	CO_2	.0200	.00245	....	....	....
Oxygen	O	.0025	.00022	....	− .009	− .0007
Nitrogen	N	.0275	.00215	....	....	....
Referred to 1 cu.ft. of gas		1.0000	.03212	571.5	5.225	.4217

Table 120 contains the main constants for illuminating gas of most interest to the designer, as computed from the average of the twelve analyses above given. Regarding the method of obtaining the figures in the various columns, Section 4, Appendix, and especially Tables 147 and 148 will give detailed information.

Average Constant for Illuminating Gas: Specific weight, .032 lb. per cu.ft.; density, .4 (air = 1.0); lower heating value, 560 B.T.U. per cu.ft., or 17 500 B.T.U. per lb., minimum air required, 5.25 cu.ft. per cu.ft., or .4217 lb. per cu.ft., or 13.2 lbs. per lb.

2. Oil Gas. Oil gas is made from cheap crude or industrial oils by vaporizing them in retorts at red heat and further heating the resulting oil vapor. The kind of oil used depends largely upon the locality where the gas is made. German oil gas installations for instance use almost entirely a paraffine oil which is a by-product of the manufacture of paraffine from bituminous coal tar; in other places suitable crude oils or petroleum residuum, when the latter can be had in sufficient quantity, are employed. 100 lbs. of gas oil will develop from 750 to 950 cu.ft. of cooled gas having a specific weight of from .043 to .058 lbs. per cu.ft. and a lower heating value varying from 900 to 1100 B.T.U. per cu.ft. The amount of coke used in heating the retorts for every 100 lbs. of oil gasified is from 70 to 80 lbs. The process therefore recovers from 30 to 40% of the heat contained in oil and coke.

The composition of oil gas depends mainly upon the kind of oil from which it is made. The principal constituents are always hydrocarbons, from which the gas also obtains its high heating value.[1] According to the firm of Julius Pintsch, Berlin, an oil gas made from residuum of petroleum gave the following volumetric analysis: 17% ethylene, 58% methane, and 24.3% hydrogen. Another oil gas, made from paraffine oil, contained 28.9 volume per cent of ethylene, 54.9% methane, 5.6% hydrogen, 8.9% carbon monoxide, and .9% carbon dioxide. Table 121 following is based upon the latter analysis, the remainder of which is not known.

TABLE 121

CONSTANTS FOR OIL GAS

Composition of Oil Gas.	1 Cubic Foot of Gas Contains		H_u B.T.U.	Air Theoretically Required	
	Cubic Feet.	Pounds.		In Cubic Feet.	In Pounds.
Hydrogen . . . . H	.056	.0031	16.63	.132	.0106
Methane . . . . CH_4	.549	.0245	522.64	5.152	.4158
Ethylene . . . . C_2H_4	.289	.0226	451.99	4.110	.3317
Carbon monoxide . . . . CO	.089	.0069	30.43	.021	.0017
Carbon dioxide . . . . CO_2	.009	.0011	. . . .	. . . .	. . . .
Referred to 1 cu.ft. of gas	.992	.0583	1021.7	9.415	.7598

Average Constants for Oil Gas: Specific weight, .058 lbs. per cu.ft.; density, .6 (air = 1.0); lower heating value 1000 B.T.U. per cu.ft. or 17 300 B.T.U. per lb.; minimum air required, 9.5 cu.ft. per cu.ft., or .76 lb. per cu.ft., or 13.2 lbs. per lb.

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[1] This type of oil gas should be distinguished from the water-oil gas made by the Lowe or similar processes.

3. Power or Producer Gas. Producer gas burns with a blue (non-illuminating) flame and is a modified water gas used mainly for power or heating purposes. It is manufactured in generators usually forming a part of the engine installation. The process consists in leading a mixture of air and steam through an incandescent layer of coke or coal, the result being that the carbon dioxide produced by the combustion is reduced to carbon monoxide, that the steam is decomposed, forming hydrogen and carbon dioxide, and finally the latter gas is also largely reduced to carbon monoxide. The first, and for some years only successful, power gas generator was developed by the English engineer Dowson,[1] from whom this gas derived the name by which it is largely known even to-day.

The most suitable fuels for generators are gas coke and coals which do not contain any easily condensible gases and as little as possible of any non-combustible ingredients. These fuels give little trouble by interfering with continuous operation and furnish a gas that may be easily cleaned. Coke and anthracite coal are mostly used, and only very recently have attempts to build generators and cleaning apparatus for bituminous coal met with any considerable measure of success.

One pound of coke or anthracite will yield from 65 to 77 cu.ft. of gas having an average heating value of from 125 to 135 B.T.U. per cu.ft. The amount of water used per lb. of fuel is generally in the ratio of 1 to 1. A water supply greater than this is not economical, considered from the standpoint of thermal efficiency (see p. 521). The temperature at which the gas leaves the generator first decreases and then increases between any two charging periods. The range of variation is between 900 and 1350° F. The density of the gas varies from .85 to .95 (air = 1.0), so that its specific weight is from .068 to .076 lb. per cu.ft. The combustion of 1 cu.ft. of this gas requires theoretically from .95 to 1.1 cu.ft. of air. Being a mixture of air-gas and water-gas, producer gas consists mainly of carbon monoxide, hydrogen, carbon dioxide and nitrogen. The composition, however, depends largely upon the existing condition in the generator, and is therefore subject to constant and considerable change. Table 122, taken from the report of a test made by E. Meyer,[2] clearly shows this.

TABLE 122

PRODUCER GAS ANALYSES

		Time of Taking Gas Sample.								Mean of the Eight Samples
		9:40	10:55	11:55	12:56	2:04	3:15	4:20	8:10	
Composition in volume, %	CO_2	6.5	4.8	4.2	4.7	5.0	4.0	4.8	4.6	4.8
	CO	26.6	28.2	27.8	26.7	26.6	29.0	28.4	27.8	27.6
	CH_4	1.3	2.7	2.6	1.8	2.2	1.7	2.0	1.6	2.0
	H	6.8	5.9	6.4	8.8	9.1	8.0	5.8	5.1	7.0
	O	.1	0	0	.1	0	0	.2	.1	0
	(diff.) N	58.7	58.4	59.0	57.8	57.1	57.3	58.8	60.9	58.6

Soon after each charging period the percentage of combustible gases always considerably increases. In some cases the gases of distillation from the fresh fuel may even double the normal heating value of the gas, especially if a considerable decrease in the volume yield of the producer is combined with the charging period. Of the total fuel used in pressure gas installations, from $\frac{1}{10}$ to $\frac{1}{6}$ is used in separate boilers for the making of the necessary steam. In suction-gas plants the heat required for this purpose is furnished by the producer itself. Through incomplete combution, direct loss of gas and radiation, from 20 to 25% of the heat developed in the producer is lost; so that from 75 to 80, in specially favorable cases up to 83%, of the heating value of the fuel is recovered in the gas. Owing to stand-by losses this efficiency is decreased to from 60–70%.

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[1] See German patent, No. 27165, 1887.

[2] Z. d. V. D. I., 1896, p. 1239 and 1304.

The mathematical determination of the amount of gas produced per pound of coal and the resulting efficiency of the gasification process is treated in Section 5 of the Appendix. Prof. E. Meyer, in the report above mentioned, used the following method of computation which, on account of the resulting simple equations for V and η_t, serves to give a rapid numerical survey of the entire process.

Assume that the chemical analysis of the gas shows the following: a vol. per cent CO_2, b vol. per cent CO, and c vol. per cent CH_4. It is also assumed that the gas contains no other hydrocarbons.

Now two volumes of CO_2 were produced from two volumes of oxygen and one volume of carbon, the latter as a gas having a specific weight twelve times that of hydrogen, that is, $12\times.00559=.067081$ lb. per cu.ft. at 32° F. and 760 mm. barometer. Hence in 1 cu.ft. of CO_2 there must be $\frac{1}{2}$ cu.ft. of carbon (considered as gas) or $\frac{1}{2}\times.06708=.03354$ lb. of carbon.

The same amount of carbon is, however, also found in 1 cu.ft. of CO and in 1 cu.ft. of CH_4, as similar computation will show. 100 cu.ft. of gas of the composition above given therefore contain $.03354\,(a+b+c)$ pounds of carbon, or 1 pound of carbon produces

$$V'=\frac{100}{.03354(a+b+c)}=\frac{2981}{a+b+c}\text{ cu.ft. of gas} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

at 32° F. and 760 mm. barometer.

The fuel, however, never consists of pure carbon. One pound of the former contains ϕ pounds of the latter, in which $\phi<1>0$ and is determined from a chemical analysis of the fuel. One pound of fuel will then produce

$$V=\frac{2981\phi}{a+b+c}\text{ cu.ft. of producer gas,} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

irrespective of what ingredients other than the carbon combinations above specified the gas may contain. If, however, the gas, besides CO_2, CO, and CH_4, also contains C_2H_4, so that the analysis shows, say d volume per cent of C_2H_4, the production of gas per pound of fuel then becomes

$$V=\frac{2981\phi}{a+b+c+2d}\text{ cu.ft.,} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2a)$$

since 1 cu.ft. of C_2H_4 contains $2\times.03354=.06708$ lb. of C.

Let H be the lower heating value of 1 cu.ft. of producer gas, and K the lower heating value of 1 lb. of the fuel. Of the latter, the part B_g lbs. is used in the generator and the part

B_d in the boiler, the ratio $\frac{B_d}{B_g}$ being assumed as known. Further, if η_t is the thermal efficiency of the entire producer plant, then

$$\eta_t = \frac{VB_gH}{K(B_g+B_d)} = \frac{VH}{\left(1+\frac{B_d}{B_g}\right)K} \qquad (3)$$

The ratio $\frac{B_d}{B_g}$ may for approximate computations be put equal to .1 for installations having a separate boiler, where no separate boiler is employed, $\frac{B_d}{B_g}=0$. In the former case we may then write

$$\eta_t = \frac{2981\phi H}{1.1(a+b+c)K}. \qquad (4)$$

If h represents the percentage of hydrogen in the gas by volume, the heating value of 1 cu.ft. of the gas produced may be expressed by

$$H=(3.42b+9.52c+2.97h) \text{ B.T.U.} \qquad (5)$$

The computation, however, will give more accurate results if H is determined by means of the Junker's calorimeter instead of by computation based upon the chemical analysis.

In the case under discussion (according to Table 122), we have that $a+b+c=34.4$, hence

$$V' = \frac{2981}{34.4} = 86.7 \text{ cu.ft.,}$$

that is, 1 lb. of carbon gasified produces 86.7 cu.ft. of producer gas.

Further, $\phi=.8772$, which makes $V=76.1$ cu.ft., i.e., 1 lb. of fuel produces 76.1 cu.ft. of gas, measured at 32° F. and 760 mm. It was also found that

$B_d=.109\,B_g$, $K=13\,000$ B.T.U. per lb. of coal;

$(H)'=134.0$ B.T.U. per cu.ft., as computed from the analysis; or,

$H=135.0$ B.T.U. per cu.ft., by calorimeter determinations.

With these figures,

$$(\eta_t)' = \frac{76.1\times 134}{1.109\times 13000} = .706$$

and

$$(\eta_t) = \frac{76.1\times 135}{1.109\times 13000} = .712.$$

The formula just established for η_t is valid to the fullest extent only when all of the fuel charged is really gasified and when no fuel particles, which are not afterward recovered are removed with the ash. If we let B_a represent that part of the fuel lost in the ash, the expression for the efficiency becomes

$$\eta_t=\frac{V(B_g-B_a)H}{K(B_g+B_d)}.$$

The constants for producer gas in the following table have been computed on the basis of the average analysis from Table 122.[1]

TABLE 123

CONSTANTS FOR COKE PRODUCER GAS

Composition of Producer Gas.	1 Cubic Foot of Gas Contains		H_u in B.T.U.	Air Theoretically Required	
	Cubic Feet.	Pounds.		In Cubic Feet.	In Pounds.
Hydrogen . . . . H. .	.070	.0004	20.8	.165	.0133
Methane . . . . CH_4	.020	.0004	19.0	.188	.0152
Carbon monoxide . . . . CO .	.276	.0215	94.2	.649	.0525
Carbon dioxide . . . . CO_2.	.048	.0058	. . . .	. . . .	. . . .
Nitrogen . . . . N . .	.586	.0458	. . . .	. . . .	. . . .
Referred to 1 cu.ft. of gas	1.000	.0744	134.0	1.002	.0810

Average Constants for Coke Producer Gas: Specific weight, .075 lb. per cu.ft.; density, .93 (air = 1.0); lower heating value, 135 B.T.U. per cu.ft., or 1800 B.T.U. per lb.; minimum air required, 1 cu.ft. per cu.ft., or .081 lb. per cu.ft., or 1.10 lbs. per lb. The above gas was made from coke having a heating value of 13 210 B.T.U. and therefore contains but little hydrogen, which manifests itself in the low heating value and high specific weight of the gas. Producer gas from anthracite always contains more hydrogen, consequently shows a higher heating value and lower weight. The difference is clear from Table 124, which gives the constants for a producer gas made from a Belgian hard coal having a heating value of 14 200 B.T.U.[2]

TABLE 124

CONSTANTS FOR ANTHRACITE PRODUCER GAS

Composition of Producer Gas.	1 Cubic Foot of Gas Contains		H_u in B.T.U.	Air Theoretically Required	
	Cubic Feet.	Pounds.		In Cubic Feet.	In Pounds.
Hydrogen . . . . H. .	.242	.0014	71.8	.572	.0462
Methane . . . . CH_4	.020	.0009	19.0	.188	.0152
Carbon monoxide . . . . CO .	.166	.0129	56.7	.390	.0314
Carbon dioxide . . . . CO_2.	.113	.0139	. . . .	. . . .	. . . .
Nitrogen . . . . N . .	.459	.0359	. . . .	. . . .	. . . .
Referred to 1 cu.ft. of gas	1.000	.0650	147.5	1.150	.0928

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[1] In conjunction with Tables 147 and 148, Appendix.

[2] Analysis of coal is given in Z. d. V. D. I., 1896, p. 350, of the gas in Z. d. V. D. I., 1895, p. 1540.

Average Constants for Anthracite Producer Gas: Specific weight, .065 lb. per cu.ft.; density, .8 (air=1.0); lower heating value 147 B.T.U. per cu.ft.=2260 B.T.U. per lb.; minimum air required, 1.15 cu.ft. per cu.ft., or .093 lb. per cu.ft., or 1.45 lbs. per lb.

An ample supply of steam to the generator is of advantage from a practical standpoint, since it tends to decrease clinkering and to prevent the rapid burning away of lining and grates. Too high a percentage of hydrogen in the gas, however, leads to heavy explosions in the cylinder of the engine. Only a few engines can stand from 7 to 10% of hydrogen in the mixture, i.e., from 15 to 20% in the producer gas; in most of them, under continued heavy load, a troublesome knocking appears as soon as the gas contains more than 10% of hydrogen. The composition of the producer gas should therefore not be made entirely dependent upon the efficiency of the gasification process (see p. 536 and 597).

4. Blast Furnace Gas. Blast furnace gas, although a gas low in heating value, is yet readily combustible, and, forming a by-product of blast furnace operation, is available nearly without cost in such quantities that one-fourth of it would be sufficient to cover the power demand of the blast furnace, leaving the other three-fourths available for other purposes. The manufacturer of 1 ton of pig iron requires the use of roughly 1 ton of coke together with about 4 tons of air (blast). The production of 1 ton of pig iron is therefore accompanied by the evolution of 5 tons, that is 140 000 cu.ft., of blast furnace gas, and a medium sized stack of say 150 tons capacity therefore furnishes about 21 000 000 cu.ft. of gas in 24 hours. Assuming that one-half of this volume is used in the hot blast stoves of the plant, this leaves 10 500 000 cu.ft. available, which quantity used in gas engines is sufficient to generate at least 4000 H.P. If used under steam boilers the same volume of gas would not produce 1500 engine H.P. Realizing that some blast furnace plants produce from 1100 to 1200 tons of pig iron every 24 hours, that is, eight times the capacity assumed, the great importance, as far as the steel works are concerned, of the direct utilization of blast furnace gas as fuel gas in engines becomes at once apparent. To Fr. W. Lürman belongs the credit of first having pointed out, in 1886, the availability of blast furnace gas for use in gas engines, and to have induced gas engine builders into experimenting with this gas.

Since the blast furnace is fundamentally nothing but an immense gas producer, it will suffice to refer to page 592 of the Appendix regarding the process of gasification. In this case, however, the charge not only consists of coke but in addition also of the burden, that is of certain quantities of iron ore and of flux, usually limestone. This modification makes both the carbon dioxide and the water vapor content of the resulting gas greater than the percentage of the corresponding constituents found in ordinary producer gas. Neglecting the large percentage of dust and water vapor carried, blast furnace gas contains about the same component gases as producer gas, that is mainly carbon monoxide and nitrogen, and, as secondary admixtures, carbon dioxide and hydrogen. The composition of the gas, owing to its manufacture as a by-product, varies considerably in different blast furnaces and at different intervals. The combustible constituents fluctuate between 25 and 35%, and the heating value correspondingly varies from 80 to 112 B.T.U. per cu.ft. The following table, published by Fritz Lürman,[1] gives some detailed information regarding this point.

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[1] Stahl und Eisen, Heft 21, 1901.

TABLE 125

ANALYSES OF BLAST-FURNACE GASES

Composition of Blast-furnace Gas.	Eisenhüttenkunde von Ledebur, 1893 p. 100.		Works in Westphalia.		Works in			Average.
	Dry. 1	Containing 10% H_2O. 2	Dry. 3	Containing 10% H_2O. 4	Upper Silesia. 5	Upper Silesia. 6	The Minette District. 7	8
Carbon monoxide.......CO	24.0	21.6	29.0	26.1	29.7	23.0	27.5	25.84
Hydrogen..............H	2.0	1.8	4.0	3.6	6.3	....	3.0	2.96
Methane...............CH_4	2.0	1.8	....	....	....	....	....	.54
Carbon monoxide.......CO_2	12.0	10.8	10.0	9.0	7.8	6.0	10.0	9.37
Nitrogen..............N	60.0	54.0	57.0	51.3	56.2	59.0	54.5	56.00
Water vapor...........H_2O	....	10.0	....	10.0	....	12.0	5.0	5.29
γ in lbs./cu.ft..............	.0814	.0782	.0795	.0767	.0799	.0775	.0790	.0789
H_u in B.T.U./cu.ft..........	105.7	95.3	109.6	98.7	100.0	77.5	101.7	98.5

The greatest, at first apparently insurmountable, obstacle that stood in the way of the successful utilization of blast-furnace gas in the gas engine was the large amount of dust carried by it. In especially unfavorable cases 1 cu.ft. of gas may carry .6 gram of dust, besides certaining metallic vapors which assume the form of dust only after combustion in the cylinder. It is comparatively easy to take out of the gas the coarse coke, iron oxide or lime stone dust, for which purpose even the simple cleaning apparatus installed in many works before the introduction of the gas engine is quite sufficient. These cleaners or washers are, however, not able to take care of the fine dust, which is of most danger to the engine, and it therefore became necessary first to invent and construct suitable apparatus. Although this important problem is as yet not completely solved, it may be said that the present-day scrubbers are capable of reducing the dust content of very dirty gas to such an extent that neither the reliability of operation or the durability of the blast furnace gas engine need any longer be seriously questioned.

To thoroughly clean the gas of the very fine dust which remains suspended in the gas current for hundreds of feet, is only possible by means of some wet method, which accordingly is the scheme most commonly adopted. After the gas has been given a preliminary dry cleaning by passing it through settling-tanks, dust-catchers, etc., it is next led through scrubbers, wet blowers or ventilators and other types of washers and, emerging from these, it is led directly to the engine or to gas holders. The large quantity of water vapor and the metallic vapor that the gas originally carries are also largely thrown down in these wet scrubbers. The water used in the latter varies from 2 to 10 cu.ft. per 1000 cu.ft. of gas cleaned. The quantity required is therefore considerable under certain circumstances, and for that reason it is in many cases recovered for future use by means of settling tanks or cooling ponds. Coke scrubbers for blast furnace gas have not been very successful. The passages between the layers of coke and between the individual pieces are soon partly or entirely stopped up by the coarse dust, and the flow of gas gradually ceases. A number of blast furnace gas cleaning installations in actual operation have been described in detail by Lürman and others, and the figures in the following table have been abstracted from these descriptions.

TABLE 126

FIGURES RELATING TO THE OPERATION OF BLAST-FURNACE GAS-CLEANING APPARATUS

	Gute Hoffnungs Hütte.	Georgs-Marien Hütte.	Friedens-Hütte O.-S.	Düdelingen.	Donnersmarkhütte	Differdingen.
1. Dust content of gas:						
(*a*) Ahead of the dry cleaner, in 1 cu.ft. of gas, grams	.167	.212	....	....	....	.302
After passing dry cleaner, in 1 cu.ft. of gas, grams	.150	....	.142	....	....	....
(*b*) Ahead of the wet scrubber, in 1 cu.ft. of gas, grams		....	....	.137	....	.151
After passing wet scrubbers, in 1 cu.ft. of gas, grams	.085	.083	.017–.045	.011	....	....
(*c*) Ahead of last cleaner, in 1 cu.ft. of gas, grams	.013	....	....	.007 behind blower	.071	.077
After passing last cleaner, in 1 cu.ft. of gas, grams	.007	....	.0006	....	.003	.011
(*d*) At the engine	....	....	....	.005	....	....
2. Water carried by gas:						
(*a*) Ahead of the cleaning app., in 1 cu.ft. of gas, grams	7 vol. %	4.02	.382	....	.850	....
(*b*) After passing cleaning app., in 1 cu.ft. of gas, grams	1.5 vol.%	.772	.156	....	.056	....
3. Temperature of gas:						
(*a*) Ahead of cleaning apparatus, ° F.	338	323	262	194–212	102	Ahead of blower 115
(*b*) After passing cleaning apparatus, ° F.	105–113	75	46–55	48–53	63	After passing blower 105
(*c*) At the engine, ° F.	air-temp.	....	....	27–53	....	73–82
4. Pressure, inches of water:						
(*a*) Ahead of cleaning apparatus, in.	5.92	....	3.56–5.92	.2–.4	.21	1.18
(*b*) After passing cleaning apparatus, in.	3.56	....	.78–2.36	1.97	1.97	3.16–3.96
5. Quantity of gas washed per hour, cu.ft.	925000 847000 for stoves 78000 for engines	353000	77500 to 38800	88000 to 53000	....	318000
6. Quantity of washing or cooling water, per hour, cu.ft.	1765	3530	....	....	762 .012 cu.ft. per cu.ft.	495
7. Quantity of dust removed per hour, lbs.	132.5	....	18.8	....	....	78.0
8. Consumption of scrubber water:						
(*a*) Per eff. H.P. hour, cu.ft.	.236	....	....	....	....	.166
(*b*) Per cubic foot of gas, cu.ft.	.067	.353	....	....	....	.055
9. Cooling or settling ponds:						
(*a*) Superficial area, sq.ft.	465	1290000	5380	....	....	12700 cu.ft.
(*b*) Depth of water, in.	43.4	59.0	....	....	....	....
(*c*) Cost of construction, dollars	23400	7500	34500	....	2700	....

It will be seen that the quantity of dust the gas may carry is extremely variable. It sometimes happens that the gas at the outset only contains from .007 to .014

gram of dust per cu.ft., in which case wet cleaning may be dispensed with altogether. According to the Societé Cockerill, Seraing, however, wet scrubbing is necessary in all cases where hæmatite, or mixtures of the same, are reduced in the blast furnace.

The following table of constants for blast furnace gas is based upon the last column of Table 125, the analysis there given having been approximated to some extent without great error.

TABLE 127

CONSTANTS FOR BLAST FURNACE GAS

Composition of Blast Furnace Gas.	1 Cubic Foot of Gas Contains		H_u in B.T.U.	Air Theoretically Required	
	Cubic Feet.	Pounds.		In Cubic Feet.	In Pounds.
Hydrogen . . . . H	.030	.0002	8.9	.071	.0057
Methane . . . . CH_4	.005	.0002	4.7	.005	.0004
Carbon monoxide . . . . CO	.260	.0203	88.8	.611	.0493
Water vapor . . . . H_2O	050	.0025	....	....	....
Carbon dioxide . . . . CO_2	.095	.0116	....	....	....
Nitrogen . . . . N	.560	.0438	....	....	....
Referred to 1 cu.ft. of gas	1.000	.0786	102.4	.687	.0554

Average Constants for Blast Furnace Gas: Specific weight, .0786 lb. per cu.ft.; density, .98 (air = 1.0); lower heating value, 101 B.T.U. per cu.ft., or 1290 B.T.U. per lb.; minimum air required, .7 cu.ft. per cu.ft., or .055 lb. per cu.ft., or .72 lb. per lb.

5. Coke Oven Gas. Coke oven gas is another of the industrial gases, being a by-product of the manufacture of coke used for metallurgical purposes (dry distillation of bituminous coal). There is fundamentally no difference between this process and the manufacture of illuminating gas. Coke oven gas therefore possesses the same heating value and in general much the same characteristics as illuminating gas. It is for these reasons a good fuel for the operation of gas engines, and, after a period of senseless waste which lasted until a very short time ago, this gas is now used in many places in gas engines with great economy. As it appears at present, the coke oven gas engine has a successful future before it.

According to the statements of Schmidt, in "Stahl und Eisen," 1901, p. 259, 1000 lbs. of bituminous coal will in coke ovens yield 3800 cu.ft. of coke oven gas, of which approximately one-third is first recovered as illuminating gas, while the remaining two-thirds are used for power and heating purposes. Table 128 is based upon the figures resulting from averaging a large number of analyses of coke oven gas, which have appeared in "Stahl und Eisen" and in the *Gas-Journal.* The heating value, according to the figures available, varies between 645 and 338 B.T.U. per cu.ft.

Average Constants for Coke Oven Gas: Specific weight, .027 lb. per cu.ft., density, .36 (air = 1.0); lower heating value 545 B.T.U. per cu.ft., or 18 700 B.T.U. per lb.; minimum air required, 5 cu.ft. per cu.ft., or .4 lb. per cu.ft., or 14.8 lbs. per lb.

TABLE 128

CONSTANTS FOR COKE OVEN GAS

Composition of Coke Oven Gas.	1 Cubic Foot of Gas Contains		H_u in B.T.U.	Air Theoretically Required	
	Cubic Feet.	Pounds.		In Cubic Feet.	In Pounds.
Hydrogen . . . . H	.550	.0031	163.0	1.299	.1045
Methane . . . . CH_4	.320	.0143	304.5	3.008	.2420
Ethylene . . . . C_2H_4	.015	.0012	23.4	.213	.0172
Benzol . . . . C_6H_6	.008	.0018	31.0	.284	.0229
Carbon monoxide . . . . CO	.070	.0055	24.0	.165	.0133
Water vapor . . . . H_2O	.010	.0005	. . . .	. . . .	. . . .
Carbon dioxide . . . . CO_2	.012	.0015	. . . .	. . . .	. . . .
Nitrogen . . . . N	.015	.0012	. . . .	. . . .	. . . .
Referred to 1 cu.ft. of gas	1.000	.0291	545.9	4.969	.3999

6. Brown Coal or Lignite Gas. This gas is apparently of no importance as far as the United States are concerned. It is one of the by-products of the manufacture of tar from brown coal, the others being coke and ammonia. Its composition is a good deal like that of coke oven gas, but it is considerably lower in heating value. 1 cu.ft. of brown coal will yield about 120 cu.ft. of gas. The latter is readily combustible in gas engines, and has been used for this purpose for years wherever available. It is not likely, however, that the gas will ever assume any great importance, since the manufacture of brown coal tar is a highly localized industry and not carried on to any extent.

The figures in the table following are based upon data given by Dammer in his "Chemical Technology."

TABLE 129

CONSTANTS FOR BROWN COAL OR LIGNITE GAS

Composition of Gas.	1 Cubic Foot of Gas Contains		H_u in B.T.U.	Air Theoretically Required	
	Cubic Feet.	Pounds.		In Cubic Feet.	In Pounds.
Hydrogen . . . . H	.243	.0014	72.0	.574	.0462
Methane . . . . CH_4	.165	.0074	156.8	1.550	.1250
Heavy hydrocarbons . . . . C_nH_{2n}	.014	.0011	27.7	.199	.0160
Carbon monoxide . . . . CO	.081	.0064	27.7	.190	.0153
Hydrogen sulphide . . . . H_2S	.011	.0010	. . . .	. . . .	. . . .
Carbon dioxide . . . . CO_2	.170	.0209	. . . .	. . . .	. . . .
Oxygen . . . . O	.031	.0028	. . . .	−.117	−.0093
Nitrogen . . . . N	.285	.0223	. . . .	. . . .	. . . .
Referred to 1 cu.ft. of gas	1.000	.0633	284.2	2.396	.1932

Average Constants for Brown Coal or Lignite Gas: Specific weight, .065 lb. per cu.ft.; density, .80 (air = 1.0), lower heating value 285 B.T.U. per cu.ft., or 4400 B.T.U. per lb.; minimum air required, 2.4 cu.ft. per cu.ft., or .20 lb. per cu.ft., or 3 lbs. per lb.

7. Acetylene. Since the commercial introduction of calcium carbide, acetylene, a colorless hydrocarbon gas (C_2H_2), has occasionally been used as a gas engine fuel. It is made from calcium carbide to which water is added at a uniform rate in simple gas producers. During the process the carbide breaks up into acetylene with the formation of calcium hydroxide as a by-product. Under favorable conditions, 1 lb. of carbide will yield 4.8 cu.ft. of acetylene. The heating value of the gas is 1480 B.T.U. per cu.ft. and the specific weight .074 lb. per cu.ft. Its density, referred to air, is therefore .91. For complete combustion this gas requires theoretically 11.85 cu.ft. per cu.ft., or .96 lb. per cu.ft., or 13.2 lbs. per lb. In practice the excess of air used should be very large to prevent the extremely severe explosions occurring with mixtures approximating the theoretical. Explosibility does not cease even with a mixture in the volume ratio of $\frac{\text{air}}{\text{gas}}=40$.

A general introduction of acetylene gas as a gas engine fuel is probably out of question on account of the high cost of carbide, the fire risks involved, and the unpleasant odor the gas possesses.

8. Natural Gas. This gas is obtained from wells in various parts of the United States. Its composition differs somewhat according to locality, and sometimes also changes from time to time in the same well. In nearly all cases, however, the main constituent is marsh gas, CH_4. The following table is based upon data given by Kent for a natural gas from Anderson, Indiana.

TABLE 130

CONSTANTS FOR NATURAL GAS

Composition of Gas.	1 Cubic Foot of Gas Contains		H_u in B.T.U.	Air Theoretically Required	
	Cubic Feet.	Pounds.		In Cubic Feet.	In Pounds.
Hydrogen . . . . H	.0186	.0001	5.5	.045	.0036
Marsh gas . . . . CH_4	.9307	.0415	885.9	8.896	.7117
Olefiant gas . . . . C_2H_4	.0047	.0004	7.4	.068	.0054
Carbon monoxide . . . . CO	.0073	.0005	2.5	.016	.0013
Carbon dioxide . . . . CO_2	.0026	.0003	. . . .	. . . .	. . . .
Oxygen . . . . O	.0042	.0004	. . . .	− .015	− .0012
Nitrogen . . . . N	.0304	.0024	. . . .	. . . .	. . . .
Hydrogen sulphide . . . . H_2S	.0015	.0001	. . . .	. . . .	. . . .
Referred to 1 cu.ft. of gas	1.000	.0457	901.3	9.000	.7208

Average Constants for Natural Gas: Specific weight, .046 lb. per cu.ft.; density, .575 (air = 1.0); lower heating value, 900 B.T.U. per cu.ft., or 19 600 B.T.U. per lb.; minimum air required, 9 cu.ft. per cu.ft. or .72 lb. per cu.ft., or 15.5 lbs. per lb.

II. Liquid Fuels

1. Crude Oil and its Distillates. The fuel oils are mainly used as fuels for the smaller sizes of internal combustion engines. Their use for larger sizes is, among other things, generally prohibited by their cost. Crude oil is a native, generally liquid, combination of various hydrocarbons, and as such usually consists of from 80 to 86%

of carbon and from 15 to 10% of hydrogen, which are the main constituents. There is usually also a small percentage of impurities. The relative proportions of the various constituents vary somewhat according to the locality where the oil is found, and also to some extent upon the oil's geologic age, as is indicated in Table 131.[1]

TABLE 131

ANALYSES OF CRUDE OIL

	Specific Gravity at 32° F.	Composition.			Heating Value B.T.U.
		Carbon, C.	Hydrogen, H.	Oxygen and Impurities.	
American petroleum	.820	83.4	14.7	1.9	17588
Heavy crude, West Virginia	.873	83.5	13.3	3.2	18324
Light crude, West Virginia	.841	84.3	14.1	1.6	18401
" Pennsylvania	.816	82.0	14.8	3.2	17933
Heavy crude, Pennsylvania	.886	84 9	13.7	1.04	19210
Crude oil, Parma	.786	84.0	13.4	1.8	18218
" Pechelbronn	.912	86.9	11.8	1.3	17474
" "	.892	85.7	12.0	2.3	18036
" Schwabweiler	.861	86.2	13.3	.5	18844
" "	.829	79.5	13.6	6.9	....
" Hannover	.892	80.4	12.7	6.9	....
" "	.955	86.2	11.4	2.4	....
" Eastern Galicia	.870	82.2	12.7	5.7	18153
" Western Galicia	.885	85.3	12.6	2.1	18415
" Balachany	.882	87.4	12.5	.1	21060
Light crude, Baku	.884	86.3	13.6	.1	20628
Heavy crude, Baku	.938	86.6	12.3	1.1	19440
Crude oil residuum, Baku	.928	87.1	11.7	1.2	19260
Crude oil, Jawo	.923	87.1	12.0	.9	19496
Heavy crude, Ogaio	.985	87.1	10.4	2.5	18146

In its native state, crude oil is a viscous liquid, having a dark brown color. Its specific gravity varies from .81 to .90, and its flash-point from 75 to 95° F. It is a mixture of several kinds of oil of differing boiling points, which during the refining process are by fractional distillation separated into groups, all the constituents of each of which have approximately the same boiling point. After the distillates have been separated there remains a heavy oil, called Masut (Ostaki) or crude oil residuum, which has a specific gravity of from .90 to .915 and a flash-point between 175 and 355° F. Crude oil as well as the residuum, on account of their high viscosity and high vaporizing temperatures, are burned only with some difficulty in internal combustion engines. The gas engine liquid fuels therefore generally belong to the class of distillates. The refiners as yet have no standard nomenclature for the various distillation groups. According to Höffer[2] the one in the appended table is perhaps the best known.

According to government regulations in Germany and Austria-Hungary, only such of the distillates as show a flash-point of at least 70° F. when tested in the

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[1] From Veit, Das Erdöl, p. 432.

[2] Höfer, Das Erdöl and seine Verwendung.

Abel instrument and referred to 760 mm. barometer, can be used as kerosene oil for lighting purposes. By *flash-point* is meant the temperature at which the oil commences to give off inflammable vapor. If these vapors are ignited, they will burn for short intervals at the surface of the oil, without igniting the body of oil. Only when the *burning-point* is reached, are the vapors generated at a rate fast enough to maintain a flame over the surface, i.e., the liquid itself burns. It follows from this that the burning-point is higher than the flash-point (by from 9 to 45°.).

The heating value of crude oil and its distillates varies from 17 000 to 21 600 B.T.U. per lb., the average being about 18 900 B.T.U. When the chemical composition of an oil is definitely known, the heating value may be computed by the combination formula, p. 591, Appendix. This formula, using the accurate figures for the heating values of C and H, may be written

$$H_u=\left[14\,544\,C+51\,840\left(H-\frac{O}{8}\right)\right] \text{ B.T.U. per lb., .} \quad (6)$$

in which C and H are expressed in parts of a pound.

Absolutely pure lighting oil is a hydrocarbon having the formula $C_{13}H_{28}$, a specific gravity of $\gamma=.793$. One pound of this oil therefore contains, using 12 as the atomic weight for C and 1 for H,

$$\frac{12\times13}{12\times13+1\times28}=.847 \text{ lbs. C,}$$

and

$$\frac{1\times28}{12\times13+1\times28}=.152 \text{ lbs. H.}$$

From this the lower heating value is

$$H_u=14\,554\times.848+51\,840\times.152=20\,232 \text{ B.T.U. per lb.}$$

Whenever possible the heating value of such oils as will readily burn in suitable burners should be determined by calorimeter.

The amount of air theoretically required for combustion is usually computed by means of the second combination formula, p. 588, Appendix. Assuming that the average composition of petroleum is .85 C, .14 H, and .01 O, this formula gives for the minimum amount of air required by 1 lb. of oil

$$L=\frac{2.667\times.85+8\times.14-.01}{.23}=14.68 \text{ lbs.,}$$

or

$$L'=\frac{2.667\times.85+8\times.14-.01}{.23\times.0807}=182 \text{ cu.ft.}$$

TABLE 132

CLASSIFICATION OF CRUDE OIL DISTILLATES

Crude Oil or Crude Naphtha. ($\gamma=.81$ to .88, Flash-point 75–95° F.)

(*a*) VOLATILE OILS

$\gamma=.65$ to .75, Flash-point below, 70° F.
Boiling-point below 300° F.

	$\gamma=$	Boiling-pt.=
Petroleum ether	.65 –.66	95–122° F.
Benzine	.66 –.68	122–158°
Gasoline (*C*-petroleum-naphtha)	.67 –.707	159–194°
Ligroin (*B*-petroleum-naphtha)	.707–.722	176–230°
Cleaning oil (*A*-petroleum-naphtha)	.722–.737	212–300°

(*b*) ILLUMINATING OILS

(Petroleum, Kerosene)
$\gamma=.78$ to .86, Flash-point 70 to 158° F.
Boiling-point above 300° F.

	γ
"Kaiser" oil	$\gamma=.78$ to .80
Ill. oil, I, American	= .80 to .81
Ill. oil, II, Russian	= .82 to .825
Standard white	= .808 to .812
Prime white	= .80 to .806
Astralin	= .85 to .86

(γ=specific gravity, water=1.0.)

(*c*) HEAVY OILS

$\gamma=.86$ to .96.
Flash-point 38¼–482° F.

	γ
Solar oils	$\gamma=.86$ to .88
Mixed oils	= .88 to .89
Cylinder oils	= .895 to .90

Etc., used only as lubricating oils.

Since the chemical composition of all crude oils and distillates varies only within very narrow limits, it may be assumed for all of them with sufficient accuracy that the minimum air required for their complete combustion is 15 lbs. per lb., or 185 cu.ft. per lb.

In determining the ratio of oil to air, it should be remembered that the former before or during the formation of the mixture changes from the state of a liquid to that of a vapor, and that its volume consequently increases considerably. The volume of vapor resulting from a given weight of oil increases with the temperature of vaporization. For any one temperature the lighter the oil, that is the more volatile constituents it contains, the greater the volume of vapor. To test the vaporizing qualities of an oil, fractional distillation is resorted to, during which the quantity of vapor evolved between certain temperature ranges is determined. Table 133 contains some detailed information on the vaporizing properties of certain crude oils. Accurate determinations of the volume of oil vapor per pound of oil are rare. One of the older numbers of the *Gas-Journal* (1895, p. 22) gives the data contained in Table 134, which originally seems to have referred to the manufacture of oil gas. Since the vaporization of oil in engine cylinders can hardly be as perfect as in special gas retorts, the figures of Table 134 will necessarily have to be reduced somewhat to apply to gas-engine practice.

TABLE 133

DISTILLATION TESTS ON MINERAL OILS

	Specific Gravity, γ	Beginning of Vaporization, ° F.	Volume Per Cent Vaporized.						
			Up to 302° F.	from 302 to 392° F.	from 392 to 482° F.	from 482 to 572° F.	from 572 to 608° F.	above 608° F.	
Alsatian "Brilliant" petroleum	.801	296	.8	30.5	44.7	20.2	3.8	....	Tests by C. Engler and C. Schestopal
"Kaiser" oil	.795	212	....	29.7	32.3	26.3	11.7	....	
Pennsylvania kerosene	.800	212	15.8	22.0	19.25	16.8	26.15		
Bavarian crude oil	.827	....	14.7	12.1	10.7	9.1	7.8	....	H_u=18250 B.T.U. Tests made by Maschinen-fabrik Augsburg
Roumanian illuminating oil	.815	....	28.2	15.2	12.8	24.3	7.7	5.4	H_u=18320 B.T.U.
Galician solar oil	.874	....	....	....	2.7	65.1	14.0	....	H_u=17750 B.T.U.
Hungarian "blue" oil	.836	....	20.1	15.0	12.6	12.2	4.3	2.4	
German "red" oil	.870	....	....	3.7	38.9	39.3	12.3	....	H_u=17470 B.T.U.
German yellow paraffin oil	.860	....	....	2.4	55.0	37.0	2.8	....	
German solar oil	.825	....	13.8	57.4	25.5	1.0	.1	....	
German benzol	.873	Up to 212° F. 68%, from 212 to 302° F. 28.9 %							

TABLE 134

QUANTITY OF VAPOR OBTAINED FROM MINERAL OILS

	B-Petrol-Naphtha, γ=.730.		Lighting Oil, γ=.807.		Heavy Oil, γ=.847.			Heavy Oil, γ=.884.		
Vaporizer temperature, ° F	1112	1562	1112	1562	1112	1472	2012	932	1112	1562
1 cu.in. of oil furnishes cubic inches of vapor	451	625	469	582	401	513	594	213	368	675
1 lb. of oil furnishes cubic feet of vapor	10.4	13.8	9.3	11.0	7.6	9.7	11.3	3.7	6.7	12.3
Residue not vaporized, %	11.4	5.1	21.4	7.5	28.5	12.2	18.0	62.3	41.5	9.4

In approximate calculations, it may be assumed, for light and medium heavy oils, that the volume increase on vaporization is in the ratio of 400 to 1 at vaporizer temperatures, while for ordinary room temperature the ratio may be taken at approximately 200 to 1. The theoretical minimum ratio by volume of oil to air is accordingly from 1:30 to 1:25. Heavy oils require a vaporizer temperature exceeding 1470° F., as Table 134 shows, in order to obtain fairly complete vaporization. This explains in a measure the difficulties encountered in attempting to use these oils in gas engines.

2. Alcohol. Alcohol is the youngest and as yet the least important of our gas engine fuels. It is produced by fermentation of grape sugar, or the latter's raw material, like potatoes or grain. Theoretically 100 parts by weight of grape sugar should yield 51 of alcohol, but the actual yield is from one-fifth to one-third less than this. Pure alcohol, absolutely free from water, called absolute alcohol, has a specific gravity of .7946, a boiling point of 172.4° F. and a molecular weight of 46. The chemical composition may be written $C_2H_5+OH=C_2H_6O$. One pound of C_2H_6O, with the usual assumptions for the atomic weight of C and H, then contains.

$$\frac{12\times 2}{46}=.522 \text{ lb. C}, \quad \frac{1\times 6}{46}=.130 \text{ lb. H}, \quad \text{and} \quad \frac{16\times 1}{46}=.348 \text{ lb. O}.$$

The air theoretically required will be, by means of the combination formula,

$$L=\frac{2.667\times .522+8\times .130-.348}{.23}\sim 9 \text{ lbs. per lb. or } \sim 112.3 \text{ cu.ft. per lb.}$$

The heating value of alcohol can not be accurately computed from its chemical composition, because the molecular grouping of the atoms C_2H_6O is not definitely known. Direct measurement by calorimeter is therefore the only satisfactory way. According to the experiments made by Thomson, the higher heating value of alcohol vapor, having a density of 1.601 (air=1.0), was 13 320 B.T.U. per lb. By subtracting from this value the heat of vaporization of the water of combustion, we find the lower heating value equal to $13320-1.17\times 1090=12045$ B.T.U. per lb. Prof. E. Meyer gives the lower heating value as $H_u=11\,664$ B.T.U. per lb., basing his computation upon the work of Favre & Silberman, who give the higher heating value at $H_0=12\,931$ B.T.U. per lb. The correction for water of combustion is taken at $1.174\times 1079=1267$ B.T.U. Meyer's figure gives as the lower heating value of 1 gallon of absolute alcohol 77 089 B.T.U.

Commercial alcohol always carries a certain quantity of water, and has therefore a greater specific gravity and a lower heating value than absolute alcohol. The degree of dilution, that is the percentage of water carried, is to-day in most cases found by means of a hydrometer, which determines the per cent by *weight* of water present on the basis of specific gravity, rather than the per cent by *volume*.[1] The standard temperature for weight per-cent determinations is 15° C.; for the volume-per cent measurement, 60° F.$=15\frac{5}{9}$° C. If the hydrometer is used at any other temperature than 15° C., the reduction to the standard may be made by use of Table 135.

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[1] In the United States, the percentage of water is usually stated in volume-per cent, thus 90% alcohol means that the alcohol contains 10% by volume of water.

Table 135

OBSERVED READING OF WEIGHT HYDROMETER

Temperature, ° C.	71	72	73	74	75	76	77	78	79	80	81	82	83	84	85	86	87	88	89	90	91	92	93
	True Strength of Alcohol in Weight Per Cent (Reduced to 15° C.)																						
−5	77.8	78.8	79.8	80.6	81.6	82.6	83.6	84.6	85.6	86.6	87.6	88.4	89.4	90.4	91.4	92.2	93.2	94	95	95.8	96.8	97.8	98.6
4	77.4	78.4	79.4	80.4	81.4	82.4	83.4	84.4	85.2	86.2	87.2	88.2	89.2	90	91	92	92.8	93.8	94.8	95.6	96.6	97.4	98.4
3	77	78	79	80	81	82	83	84	85	86	87	87.8	88.8	89.8	90.8	91.6	92.6	93.6	94.4	95.4	96.2	97.2	98
2	76.8	77.8	78.8	79.8	80.8	81.8	82.8	83.6	84.6	85.6	86.6	87.6	88.6	89.4	90.4	91.4	92.4	93.2	94.2	95	96	96.8	97.8
1	76.4	77.4	78.4	79.4	80.4	81.4	82.4	83.4	84.4	85.4	86.4	87.2	88.2	89.2	90.2	91	92	93	93.8	94.8	95.8	96.6	97.6
0	76.2	77	78	79	80	82	83	83	84	85	86	87	88	88.8	89.8	90.8	91.8	92.6	93.6	94.6	95.4	96.4	97.2
+1	75.8	76.8	77.8	78.8	79.8	80.8	81.8	82.8	83.8	84.6	85.6	86.6	87.6	88.6	89.6	90.4	91.4	92.4	93.2	94.2	95.2	96	97
2	75.4	76.4	77.4	78.4	79.4	80.4	81.4	82.4	83.4	84.4	85.4	86.4	87.2	88.2	89.2	90.2	91.2	92	93	94	94.8	95.8	96.8
3	75.2	76	77.2	78	79	80	81	82	83	84	85	86	87	88	88.8	89.8	90.8	91.8	92.8	93.6	94.6	95.6	96.4
4	74.8	75.8	76.8	77.8	78.8	79.8	80.8	81.8	82.8	83.6	84.6	85.6	86.6	87.6	88.6	89.6	90.6	91.4	92.4	93.4	94.2	95.2	96.2
5	74.4	75.4	76.4	77.4	78.4	79.4	80.4	81.4	82.4	83.4	84.4	85.4	86.4	87.2	88.2	89.2	90.2	91.2	92.2	93	94	95	96
6	74	75	76	77	78	79	80	81	82	83	84	85	86	87	88	89	89.8	90.8	91.8	92.8	93.8	94.6	95.6
7	73.8	74.8	75.8	76.8	77.8	78.8	79.8	80.8	81.8	82.8	83.6	84.6	85.6	86.6	87.6	88.6	89.6	90.6	91.6	92.4	93.4	94.4	95.4
8	73.4	74.4	75.4	76.4	77.4	78.4	79.4	80.4	81.4	82.4	83.4	84.4	85.4	86.4	87.4	88.2	89.2	90.2	91.2	92.2	93.2	94	95
9	73	74	75	76	77	78	79	80	81	82	83	84	85	86	87	88	89	90	91	91.8	92.8	93.8	94.8
10	72.8	73.8	74.8	75.8	76.8	77.8	78.8	79.8	80.8	81.8	82.8	83.8	84.6	85.6	86.6	87.6	88.6	89.6	90.6	91.6	92.6	93.6	94.4
11	72.4	73.4	74.4	75.4	76.4	77.4	78.4	79.4	80.4	81.4	82.4	83.4	84.4	85.4	86.4	87.4	88.4	89.4	90.2	91.2	92.2	93.2	94.2
12	72	73	74	75	76	77	78	79	80	81	82	83	84	85	86	87	88	89	90	91	92	93	94
13	71.6	72.6	73.6	74.6	75.6	76.6	77.6	78.6	79.6	80.6	81.6	82.6	83.6	84.6	85.6	86.6	87.6	88.6	89.6	90.6	91.6	92.6	93.6
14	71.4	72.4	73.4	74.4	75.4	76.4	77.4	78.4	79.4	80.4	81.4	82.4	83.4	84.4	85.4	86.4	87.4	88.4	89.4	90.4	91.4	92.4	93.4
15	71	72	73	74	75	76	77	78	79	80	81	82	83	84	85	86	87	88	89	90	91	92	93
16	70.6	71.6	72.6	73.6	74.6	75.6	76.6	77.6	78.6	79.6	80.6	81.6	82.6	83.6	84.6	85.6	86.6	87.6	88.6	89.6	90.6	91.6	92.8
17	70.4	71.4	72.4	73.4	74.4	75.4	76.4	77.4	78.4	79.4	80.4	81.4	82.4	83.4	84.4	85.4	86.4	87.4	88.4	89.4	90.4	91.4	92.4
18	70	71	72	73	74	75	76	77	78	79	80	81	82	83	84	85	86	87	88	89	90	91	92
19	69.6	70.6	71.6	72.6	73.6	74.6	75.6	76.6	77.6	78.6	79.6	80.6	81.6	82.6	83.6	84.6	85.6	86.6	87.6	88.8	89.8	90.8	91.8
20	69.2	70.2	71.2	72.2	73.2	74.2	75.2	76.2	77.2	78.2	79.2	80.2	81.2	82.4	83.4	84.4	85.4	86.4	87.4	88.4	89.4	90.4	91.4
21	69	70	71	72	73	74	75	76	77	78	79	80	81	82	83	84	85	86	87	88	89	90.2	91.2
22	68.6	69.6	70.6	71.6	72.6	73.6	74.6	75.6	76.6	77.6	78.6	79.6	80.6	81.6	82.6	83.6	84.6	85.6	86.8	87.8	88.8	89.8	90.8
23	68.2	69.2	70.2	71.2	72.2	73.2	74.2	75.2	76.2	77.2	78.2	79.2	80.2	81.2	82.2	83.4	84.4	85.4	86.4	87.4	88.4	89.4	90.6
24	67.8	68.8	69.8	70.8	71.8	72.8	73.8	74.8	75.8	76.8	77.8	78.8	79.8	81	82	83	84	85	86	87	88.2	89.2	90.2
25	67.4	68.4	69.4	70.4	71.4	72.4	73.4	74.6	75.6	76.6	77.6	78.6	79.6	80.6	81.6	82.6	83.6	84.6	85.6	86.8	87.8	88.8	89.8

The conversion from volume-per cent (*Rtl*) to weight-per cent (*Gtl*) may be made by the formula

$$Gtl=\frac{Rtl\times .7946}{\gamma}, \quad \ldots \ldots \ldots \ldots \quad (7)$$

in which γ represents the specific gravity of the sample. The heating value of an x weight-per cent alcohol or a y volume-per cent alcohol may be approximately expressed by

$$H_u=11664\,x \text{ B.T.U. per lb.,} \quad . \quad . \quad (8) \quad \text{or} \quad H_u'=77089y \text{ B.T.U. per gallon.} \quad . \quad . \quad (8a)$$

An accurate computation of the heating value of commercial alcohol is not possible for the reasons already stated for absolute alcohol. In this case the computation is rendered still more uncertain by the fact that commercial alcohol often contains denaturizing agents. The calorimeter is therefore again the only reliable means of determining the heating value.

On account of the high specific cost of heat of alcohol, see Table 118, p. 513, the cost of operating gas engines is considerably higher with alcohol than with either gasoline or kerosene, other things being the same. In order to improve the heating value and at the same time to bring down the heat cost, alcohol is sometimes carbureted with from 10 to 50% of benzol, especially for operation in automobiles. Although this undoubtedly brings down the operating cost, the use of benzol is accompanied by certain undesirable features.[1]

Liquid benzol has a specific gravity of $\gamma=.866$, a heating value of $H_u=17\,190$ B.T.U. per lb., or $H_u'=123\,790$ B.T.U. per gallon. From this, for a mixture consisting of $a\%$ of alcohol and $b\%$ of benzol, the approximate heating value will be

$$H_u=[(a\times 11664)+(b\times 17190)] \text{ B.T.U. per lb.,}$$

or

$$H_u'=[(a\times 77089)+(b\times 123\,790)] \text{ B.T.U. per gallon.}$$

If the alcohol in the mixture is not absolute, but contains a certain per cent of water, the corrected heating value must be used in the above equations.

Mixtures of alcohol and benzol have not been tried to any extent in either stationary engines or locomobiles. The tendency here has rather been to make the alcohol engine as efficient as the gasoline or the kerosene engine by the use of higher compression. The water generally carried by commercial alcohol makes it possible to employ much higher compression pressures than it is feasible to use with crude oil or its distillates, and the efficiency of the combustion is correspondingly increased. It is possible in this way to largely overbalance the greater heat cost of alcohol by higher thermal efficiencies.

It has been observed that the interior surfaces of alcohol engines are in some cases strongly attacked by rust after a short period of operation. The burned gases must therefore contain acid constituents. These may, during the first oxidation periods, consist of aldehyde or acetic acid, forming when alcohol is burned with very little or no excess of air. With an ample air supply even the formation of acetic acid no longer takes place, the alcohol being directly oxidized to water and carbon dioxide, neither of which have any harmful effect on iron.

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[1] See Authors' Report in the "Motorwagen," p. 271.

This serves to show that alcohol engines should be built with a view to using as large an excess of air as possible.

III. Fuel Mixtures

The gas engine fuels, in preparing them for combustion in the internal combustion engines, are mixed with an amount of atmospheric air which makes the oxygen present much in excess of what is required in theory.

Pure dry air is a mechanical mixture containing

$$\left.\begin{array}{r}23.58 \text{ weight-per cent}\\ 21.33 \text{ volume-per cent}\end{array}\right\} \text{of oxygen and} \left\{\begin{array}{l}76.42 \text{ weight-per cent}\\ 78.67 \text{ volume-per cent}\end{array}\right\} \text{of nitrogen,}$$

together with traces of water vapor and carbon dioxide. It therefore requires

4.241 parts by weight of air to obtain 1 part by weight of oxygen

and 4.688 parts by volume of air to obtain 1 part by volume of oxygen.

The weight of air is .0806 lb. per cu.ft. under standard conditions of 32° F. and 29.92″ Hg (=760 mm.), or .078 lb. per cu.ft. when referred to 32° F. and the standard metric atmosphere of 735.5 mm. Hg. The specific volume, that is, the volume of 1 lb. of air consequently is 12.41 cu.ft., or 12.82 cu.ft., depending upon what are considered standard conditions. Under the average conditions of humidity, air, on account of the water vapor carried, is always lighter than above stated. Thus the average weight is found to be .0802 lb. per cu.ft. at 32° F. and 760 mm. Hg., and .0777 lb. per cu.ft. at 32° F. and 735.5 mm. Hg. The corresponding specific volumes are then increased to 12.47 and 12.87 cu.ft. per lb. respectively. To determine the specific volumes V_0 for any other temperatures (t) and other barometer readings (b), Table 136 may be used.

The *lower* as well as the *upper limits* of the allowable ratio between air and fuel are given, the former pretty definitely by the theoretical amount of air required by the given fuel, while the latter is subject to many influences and therefore difficult of exact determination. There is little doubt, however, that the field lying between the lower and upper explosive limits, that is, the so-called "explosive range," is considerably wider for similar air-fuel ratios in actual practice than has been found in laboratory tests or similar trials. The reason for this is the extension of the upper limit due to compressing and heating the mixture. Such tests, however, even if they are carried through in the usual way with volumes of mixture under ordinary pressures and temperatures, give a lot of information, valuable to the engine builder, in relation to the explosibility of the various fuel mixtures under varying degrees of dilution. The following has been abstracted from the more recent test reports.

The explosion limits recorded in Table 137 were determined by Dr. Eitner in the laboratory of the technical high school at Karlsruhe.[1] The figures given represent

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[1] Journal für Gasbeleuchtung, 1902, p. 1.

TABLE 136

SPECIFIC VOLUME V_0 (CUBIC FEET PER POUND) OF ATMOSPHERIC AIR UNDER CONDITIONS OF AVERAGE HUMIDITY

b. Mercury Column.		TEMPERATURE t=									
Inches.	Millimeters.	°F. 20 °C. −6.6	25 −3.9	32 0	40 +4.4	50 10.0	60 15.5	70 21.1	80 26.6	90 32.2	100 37.7
31.0	787.4	11.75	11.87	12.04	12.23	12.48	12.72	12.97	13.22	13.46	13.70
30.8	782.3	11.82	11.95	12.12	12.31	12.56	12.78	13.05	13.31	13.55	13.79
30.6	777.2	11.90	12.02	12.19	12.39	12.64	12.88	13.14	13.39	13.63	13.88
30.4	772.2	11.98	12.11	12.28	12.48	12.73	12.97	13.23	13.49	13.73	13.98
30.2	767.1	12.06	12.19	12.36	12.56	12.81	13.05	13.31	13.57	13.82	14.07
30.0	762.0	12.14	12.27	12.44	12.64	12.90	13.15	13.41	13.67	13.91	14.17
29.8	756.9	12.22	12.35	12.53	12.72	12.98	13.23	13.49	13.75	14.00	14.25
29.6	751.8	12.31	12.43	12.61	12.81	13.07	13.32	13.59	13.85	14.10	14.35
29.4	746.8	12.39	12.52	12.70	12.90	13.16	13.41	13.68	13.94	14.20	14.45
29.2	741.7	12.48	12.60	12.78	12.98	13.25	13.51	13.77	14.03	14.29	14.55
29.0	736.6	12.57	12.69	12.88	13.07	13.34	13.60	13.86	14.13	14.39	14.65
28.8	731.5	12.65	12.78	12.96	13.16	13.43	13.70	13.96	14.23	14.49	14.75
28.6	726.4	12.74	12.88	13.05	13.26	13.53	13.80	14.07	14.34	14.59	14.86
28.4	721.4	12.83	12.96	13.14	13.35	13.63	13.90	14.17	14.44	14.70	14.97
28.2	716.3	12.92	13.05	13.24	13.45	13.72	13.98	14.26	14.53	14.80	15.06
28.0	711.2	13.01	13.14	13.33	13.54	13.82	14.09	14.36	14.64	14.90	15.17
27.8	706.1	13.10	13.23	13.42	13.64	13.91	14.18	14.46	14.74	15.00	15.28
27.6	701.0	13.20	13.34	13.53	13.75	14.02	14.28	14.57	14.85	15.12	15.40
27.4	696.0	13.29	13.44	13.62	13.85	14.12	14.38	14.67	14.96	15.23	15.51
27.2	690.9	13.39	13.53	13.72	13.94	14.23	14.50	14.79	15.08	15.34	15.63
27.0	685.8	13.49	13.63	13.82	14.04	14.33	14.60	14.89	15.18	15.45	15.73
26.8	680.7	13.59	13.73	13.93	14.14	14.43	14.71	15.00	15.29	15.57	15.85
26.6	675.6	13.70	13.84	14.04	14.25	14.54	14.83	15.11	15.40	15.69	15.97
26.4	670.6	13.81	13.94	14.15	14.36	14.66	14.94	15.23	15.52	15.81	16.10
26.2	665.5	13.90	14.05	14.25	14.57	14.76	15.04	15.34	15.64	15.93	16.22

TABLE 137

EXPLOSIVE RANGES OF PURE AIR-FUEL MIXTURE

	Carbon Monoxide.	Hydrogen.	Water gas.	Acetylene.	Illuminating Gas.	Ethylene.	Alcohol, 95.14 Weight, %.	Methane.	Ether.	Benzol.	Pentane.	Gasoline.
Upper explosive limit (1).	16.5	9.45	12.4	3.35	7.9	4.1	3.95	6.1	2.75	2.65	2.4	2.4
Ratio $\frac{\text{air}}{\text{gas}}$ by volume . . .	5.1	9.6	7.1	28.6	11.7	23.4	24.3	15.4	35.4	36.7	40.7	40.7
Lower explosive limit (3)	74.95	66.4	66.75	52.3	19.1	14.6	13.65	12.8	7.7	6.5	4.9	4.9
Ratio $\frac{\text{air}}{\text{gas}}$ by volume . . .	.33	.51	.6	.91	4.24	5.85	6.33	6.81	11.99	14.38	19.41	19.41

volume-per cent, lines 1 and 3 showing the per cent by volume of the total volume of the mixture represented by the fuel gas. It will be noted that the first four gases have by far the greatest explosive ranges, after that comes illuminating gas, whose range is only $\frac{19.1-7.9}{74.95-16.5}=\frac{11.2}{58.45}=\text{appr.}\ \frac{1}{5}$ that of carbon monoxide. The narrowest range is shown by the hydrocarbons. The volume of air at the lower limits is considerably smaller than the theoretical amount required for combustion of the fuel concerned. The reason for this of course is that "ignitability" does not necessarily cease when this theoretical limit is reached.

Several years ago a series of experiments were made at the same laboratory by Roszkowski under the direction of Professor Bunte to determine the effect of change of temperature and of contamination of mixture by carbon dioxide upon the explosion limits. From this extensive report[1] Table 138, showing the most important results bearing directly upon the subject under discussion, has been abstracted. The carbon dioxide-oxygen mixtures always contained, besides the fuel gas, these two gases in the ratio of 79 to 21, that is, in the exact proportion that nitrogen bears to oxygen in ordinary air.

TABLE 138

EXPLOSIVE RANGES OF CONTAMINATED FUEL GAS-AIR MIXTURES

(FIGURES SHOW THE VOLUME-PER CENT OF FUEL GAS IN MIXTURE)

Mixture Consists of	Temp. 59° F.		212		392		572	
	Upper	Lower	Upper	Lower	Upper	Lower	Upper	Lower
	Limit.		Limit.		Limit.		Limit.	
Hydrogen and air	9.5	64.7	9.5	68.2	9.6	72.1	9.6	79.3
Hydrogen and [CO_2-O mixture]	13.1	68.1	11.7	69.1	12.8	65.1	14.1	61.1
Carbon monoxide and air	14.3	74.6	13.2	77.2	12.5	80.4	21.0	57.4
Carbon monoxide and [CO_2-O mixture]	21.9	72.5	20.2	74.8	26.0	70.0	38.2	62.9
Methane and air	6.8	13.0	5.8	12.6	5.8	12.8	5.7	13.0
Methane and [CO_2-O mixture]	9.0	11.6	8.7	12.0	8.7	12.4	8.5	12.2
Illuminating gas and air	7.0	22.6	7.0	24.7	6.5	26.7	6.5	28.6
Illuminating gas and [CO_2-O mixture.]	8.7	24.2	8.0	26.5	9.1	21.7	9.4	18.0

Looking at the explosion limits as determined by this table a little more in detail, we note the following:

(*a*) *Hydrogen and Air Mixture.* An increase of temperature has no effect upon the upper limit. The lower limit, on the other hand, is extended 14.6% by heating from 59 to 572°.

(*b*) *Hydrogen and Carbon Dioxide-Oxygen Mixture.* The explosive limit is narrowed more strongly by carbon dioxide than by the diluting medium, nitrogen, ordinarily used. Raising the temperature has a beneficial effect only up to 212°, beyond this point heating seems to narrow rather than to extend the explosive range.

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[1] Journal für Gasbeleuchtung, 1880, p. 491.

(*c*) *Carbon Monoxide and Air.* The lower limit is extended 5.8% by heating to 392°; at 572° the range has narrowed at the lower and upper limits 6.7 and 17.2% respectively.

(*d*) *Carbon Monoxide and Carbon Dioxide-Oxygen Mixture.* The explosive range is in this case also considerable narrower than for the carbon monoxide-air mixtures.

The upper limit is extended 1.7% at 212°, the lower limit 2.3% at the same temperature, but beyond this the narrowing of the range is marked, amounting to 4.10 and 2.5% respectively at 392°, and 16.3 and 9.6% respectively at 572°.

(*e*) *Methane and Air Mixture.* Increase of temperature has no effect.

(*f*) *Methane and Carbon Dioxide-Oxygen Mixture.* The addition of carbon dioxide has the same harmful effect as in the previous cases. Temperature rise again without effect.

(*g*) *Illuminating Gas and Air Mixture.* The temperature affects the lower limit only, extending it about 2% for every 180°.

(*h*) *Illuminating Gas and Carbon Dioxide-Oxygen Mixture.* Again the effect of adding carbon dioxide narrows the range both above and below. The temperature, raised to 572°, raises the upper limit .7% and lowers the lower limit 6.2 volume-per cent.

The addition of carbon dioxide therefore has a harmful effect in all cases, and this conclusion naturally would hold also in case the contamination of the mixture is produced by the addition of burned gases remaining in the cylinder. Raising the temperature is of marked advantage only for hydrogen and illuminating gas mixtures, which advantage, however, is again lost by the addition of carbon dioxide.

Lately Le Chatelier and Boudouard have found that very lean carbon monoxide-air mixtures are rendered explosive by rapid heating. While at ordinary temperature 16% of carbon monoxide formed the upper limit, this was changed to 14.2% at 752°, 9.3% at 912°, and 7.4% at 1112° F.[1] This determination is of importance in the utilization of producer and blast furnace gas (both of which are high in CO), especially since the explosibility of such carbon-monoxide mixtures is in itself raised by compression in the cylinder.

Concerning the influence of higher pressure upon the explosive range we know next to nothing. Only this much is certain, that a variation of pressure has no effect upon some of the fuel mixtures. Thus, for instance, Eitner and Bucerius could not find a change in the explosive limits of hydrogen-air mixtures by varying the pressure from 7 to 60 lbs., while lean carbon monoxide mixtures could be ignited with greater ease when compressed.[2]

The most important combustibles in producer gas proper are carbon monoxide and hydrogen. Both, as far as combustion in engines are concerned, have good and bad points, and the aim should be to balance these as far as possible in the producer gas itself, since in the gas-air mixture the undesirable properties of each combustible can only be slightly toned down. Concerning the general question as to what combustible gases, and how much of each, are desirable in the mixture, Kutzbach in one of his lectures makes the following statements:[3]

"Hydrogen (H) as compared to carbon monoxide (CO) has the following characteristics: 1. Its ignition temperature is considerably below that of CO. 2. Its velocity of flame propagation is at atmospheric pressure about thirty times that of CO. 3. Its diffusion velocity is much greater than that of CO, so that it mixes much more rapidly with the required air. 4. It can stand a much greater excess of air for combustion than CO. The presence of hydrogen in the mixture therefore largely determines the qualities of the combustion and also the allowable, although local, wall temperature in the cylinder. On the other hand, small variations in the hydrogen content have marked influence upon the rapidity of flame or pressure propagation through the charge and consequently also upon engine capacity and fuel consumption.

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[1] Comptes rendus, 1896, p. 1.

[2] Journal für Gasbeleuchtung, 1901, p. 838.

[3] Z. d. V. D. I., 1905, p. 238.

From the standpoint of the engine builder, therefore, *the mixture should not contain too much* H *to keep the engine from being too sensitive, nor should it contain too little, in order to help the somewhat sluggish combustion of* CO, *with its tendency to after-burning, and to accelerate the ignition throughout the mass.* At present from 10–15% of H in the producer gas, corresponding to about 5% in the mixture, seems to be about the best proportion."

Other information bearing upon this same question may be obtained from the experiments made by A. Wagener upon the determination of the ratio between air and blast furnace gas and upon which he reported in the lecture mentioned on p. 45:

"Of all the fuel mixtures tried that containing gas and air in the ratio of 4 to 3 showed the most complete combustion. This mixture burned instantaneously with the development of a red-blue flame. An analysis of the burned gases showed combustible remaining to the extent of only .7% CO. As the proportion of gas in the mixture was increased or decreased from this ratio, the combustion became less and less rapid, until, at the explosion limit, it burned only very slowly with clear blue flame. When the ratios were gas to air=4:6 and 4:1.5, no ignition could be produced even when the pressure in the burette was raised by several millimeters of mercury. The blast-furnace gas used in these experiments had the following composition: 1.5% H, 31% CO, and 9% CO_2, the remainder being N, etc.; it was burned in open burners, that is, not under pressure.

These tests also brought out the point, which, in view of Bunsen's statements regarding pure carbon monoxide, can hardly seem strange, that when blast-furnace gas is to be burned in an open burner, such as is used in the calorimeter, the exit velocity must be kept very low. Experiments made later with gas from Hörde on this point show that the critical velocity is about 2.82 ft. per sec. At this velocity the flame burns unsteadily, is lifted away here and there from the mouth of the burner, and now and then goes out altogether."

The German technical expression for mixtures consisting of a combustible gas with just sufficient oxygen for its combustion is "Knall-gas," while if the mixture consists of the combustible gas with just sufficient air the term "Luftknall-gas" is used. In English practice the term "true explosive mixture" has been used indiscriminately for either one of these combinations. The following table, due to Professor Bunte,[1] shows the volume relation between the combustible gases given and the volume of oxygen required for the true explosive mixture.

Kind of Gas.	Ratio Gas to Oxygen.
Carbon monoxide, hydrogen, and water gas	1 to ½
Methane	1 to 2
Acetylene	1 to 2½
Ethylene	1 to 3
Alcohol vapor	1 to 3
Ether	1 to 6
Benzol	1 to 7½
Gasoline (pentane)	1 to 8
Illuminating gas	about 1 to 1.2

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[1] Journal für Gasbeleuchtung, 1901, p. 837.

Multiplying the right member of each one of these ratios by 4.7, we obtain the theoretical volume of air for the true explosive mixture.

Generally the excess of air in the fuel mixture should be made as large as possible in order to draw down combustion temperatures, prevent pre-ignition, and to furnish sufficient oxygen to the fuel should the mixture be imperfect. (Concerning the thermal advantages of lean mixture, see p. 13). The upper limit, that is the greatest degree of dilution, is of course set by the failure of the mixture to ignite explosively. In actual practice mixtures are kept well below this limit in order to avoid combinations that are too lean and would consequently result in low engine capacity. The average excess of air at normal load is usually from 30 to 40%. For the rich fuels even more air than this is employed, the mixtures in general having a heating value of from 45 to 60 B.T.U. per cu.ft. (See Part II, p. 73.) The variation of this heating value for various excess coefficients and for the average composition of the gaseous or liquid fuels discussed in the previous paragraphs is shown in Table 139.

TABLE 139

WEIGHT AND HEATING VALUE OF AIR-FUEL GAS MIXTURES

The bracketed data in the column headings refer to the liquid fuels.	Specific Weight in Lbs. per Cu.Ft. (Lbs. per Gallon.)	Density, Air = 1.0.	Specific Volume in Cu.Ft. per Lb. (Gallons per Lb.)	Lower Heating Value, H_u.		Air Required for True Explosive Mixture.		Heating Value in B.T.U per Cubic Foot with Excess of Air Amounting to						
				Per Cubic Foot (per gallon)	Per Pound	Cubic Feet per Cubic Foot.	Pounds per Pound.	0	25%	50%	75%	100%	125%	150%
				of the Fuel.										
				B.T.U.	B.T.U.									
Illuminating gas . . .	.032	.40	31.3	560	17400	5.25	13.2	90	74	63.3	55	48.8	43.8	40
Oil gas	.056	.60	17.9	1010	18000	9.50	14.0	96.5	78.5	66.5	57.3	50.5	45	40.5
Prod. gas (coke)	.075	.93	13.4	135	1800	1.00	1.1	66.5	59	52.7	47.7	45	40.5	38.2
Prod. gas (anthr.) . .	.066	.80	15.1	146	2250	1.15	1.45	68	59.5	53.3	48.5	45.5	41	38.2
Blast-furnace gas . . .	.078	.98	12.8	101	1320	.70	.72	59.5	53.8	49.3	45.5	42	39.3	37
Coke oven gas	.029	.36	34.5	545	18000	5.00	14.0	91	75	59	56	49.5	48.3	44
Acetylene	.073	.91	13.7	1460	19700	11.9	13.2	113	92	77.5	66.7	59	52.7	47.7
Crude oil and residuum	7.22	. . .	.138	130000	18000	. . .	appr. 15.0	70	56	47.7	41	36.5	32.5	29.2
Kerosene	6.72	. . .	.149	126000	18800	. . .		73.5	59.5	50.5	43.2	38.2	33.7	30.3
Gasoline	5.80	. . .	.173	109000	18800	. . .		73.5	59.5	50.5	43.2	38.2	33.7	30.3
Alcohol (abs.)	6.67	. . .	.150	77700	11660	. . .	appr. 9.0	75	61.2	51.6	45	40	35.5	32
Alcohol (90 vol. %) .	6.90	. . .	.145	69000	10000	. . .		64.5	52.2	44.3	38.2	33.6	30.3	27.5
Alcohol (85 vol. %) .	7.00	. . .	.143	64700	9250	. . .		60	48.8	41	35.4	31.4	28	25.8

B. COMBUSTION IN THE GAS ENGINE

I. Theoretical Data

For the introduction to Thermochemistry, see p. 580, Appendix. Upon the basis of the principles there discussed it becomes necessary first to determine the influence of combustion upon several of the main factors occurring in constructive and thermal problems relating to the gas engine. As an example we will take the average illuminating gas for which the data is given in Table 119, p. 515, and for which the final results in Table 140, following, show an apparent molecular weight of $m=11.543$, a specific heat of $c_p=.695$, or $c_v=.523$, also a value of $x=1.332$ and a gas constant R equal to 134.8.

TABLE 140

THEORETICAL THERMAL DATA RELATING TO AN AVERAGE ILLUMINATING GAS

	Composition in		From Table 143, p. 567 in Appendix.				$V \cdot m$	$G \cdot c_p$	$G \cdot c_v$	$G \cdot R$
	Volume, Per Cent. V	Weight, Per Cent. G	m	c_p	c_v	R				
H	.485	.0845	2	3.430	2.430	775.6	.970	.2890	.2050	65.53
CH_4	.350	.4855	16	.593	.468	97.25	5.620	.2880	.2270	47.10
CO	.070	.1703	28	.245	.174	55.23	1.960	.0416	.0296	9.40
C_2H_4	.045	.1093	28	.400	.330	55.23	1.262	.0437	.0360	6.03
CO_2	.020	.0765	44	.200	.155	35.01	.880	.0153	.0119	2.67
O	.0025	.0070	32	.217	.153	49.79	.080	.0015	.0019	.35
N	.0275	.0670	28	.245	.174	55.23	.771	.0164	.0117	3.70
							11.543 $=m$	.6955 $=c_p$	.5231 $=c_v$	134.78 $=R$

The constant R may also be determined in a simpler way from

$$R_0 : R = m : m_0 = \gamma : \gamma_0 = \delta : \delta_0,$$

in which $R_0=775.6$, $m_0=2$, $\gamma_0=.00559$, and $\delta_0=.0693$ respectively, represent the gas constant, the molecular weight, the weight per cubic foot, and the density (air $=1.0$) of hydrogen. From this we may write for the given illuminating gas,

$$R=\frac{2\times 775.6}{11.55}=\frac{.00559\times 775.6}{.0321}=\frac{.0693\times 775}{.40}\sim 134.$$

Again, from the difference in the specific heats,

$$R=\frac{c_p-c_v}{A}=778(.695-.523)=778\times .172\sim 134.$$

The following table shows the constants for the air-gas mixtures, the ratios being as indicated in the first two lines.

Ratio $\frac{\text{air}}{\text{gas}}$ by volume,	$V=6$	8	10	12
Ratio $\frac{\text{air}}{\text{gas}}$ by weight,	$G=15$	20	25	30
Weight per cu.ft. of mixture, lbs.,	$\gamma=$ .0732	.0748	.0757	.0769
$R=\frac{134+53.57\,G}{G+1}$	$=58.59$	57.40	56.66	56.16
$c_p=\frac{.695+.238\,G}{G+1}$	$=$ .2667	.2595	.2556	.2527
$c_v=\frac{.523+.169\,G}{G+1}$	$=$ .1914	.1858	.1828	.1803
$x=c_p\div c_v$	$=$ 1.393	1.397	1.399	1.402

It appears from this that, as the ratio of air to gas in the fuel mixture increases, there is a small decrease in the value of R, c_p, and c_v, while x slightly increases.

The rearrangement of the gases incident to the combustion causes a change in the values of m, R, c_p, c_v and x. Table 141 shows the resulting figures, the composition of the burned gases being found by means of equations (18) to (20), p. 590, Appendix.

TABLE 141

THEORETICAL THERMAL DATA RELATING TO BURNED GASES

Volume ratio: $\frac{\text{Air}}{\text{Fuel gas}}$		6	8	10	12
Composition of burned gases volumes	Carbon dioxide, CO_2	.530	.530	.530	.530
	Water vapor, H_2O	1.275	1.275	1.275	1.275
	Oxygen, O	.150	.570	1.000	1.410
	Nitrogen, N	4.768	6.348	7.928	9.508
Volume of burned gas, per cubic foot of fuel gas	V_r, cu.ft.	6.732	8.723	10.733	12.723
Volume of mixture before combustion, per cubic foot of fuel gas	V_g, cu.ft.	7.000	9.00	11.00	13.00
Ratio: V_r to V_g		.960	.970	.976	.9785
Contraction, ΔV_g, %		4	3	2.4	2.15
Main constants for the burned gases	$R_r=R\frac{V}{V_g}$	56.3	55.8	55.4	55.0
	$\mathbf{C}_p$ (based on 1 cu.ft.)	.0204	.0200	.0199	.0197
	$\mathbf{C}_v$ (" ")	.0149	.0145	.0144	.0142
	c_p (" 1 lb.)	.2787	.2674	.2623	.2566
	c_v (" ")	.2035	.1940	.1894	.1851
	$x=\frac{c_p}{c_v}=\frac{\mathbf{C}_p}{\mathbf{C}_v}$	1.370	1.380	1.385	1.387

The computation of the specific heats of the burned gas is made by means of equations (11) and (11*a*), p. 568. The method of doing this is shown in the following table for the mixture whose ratio is 1 of gas to 6 of air by volume.

The specific heats of the typical gases are given in Table 143, p. 567, based upon unit volume. With the aid of these we will have for exhaust gas of composition shown in the first column of Table 141 of the following:

$$\begin{array}{ll|l}
\text{For } CO_2, & \mathbf{C}_p = .530 \times .0245 = .0130 & \mathbf{C}_v = .530 \times .0189 = .0100 \\
H_2O, & \mathbf{C}_p = 1.275 \times .0241 = .0307 & \mathbf{C}_v = 1.275 \times .0185 = .0236 \\
O, & \mathbf{C}_p = .150 \times .0191 = .0028 & \mathbf{C}_v = .150 \times .0136 = .0021 \\
N, & \mathbf{C}_p = 4.768 \times .0191 = .0910 & \mathbf{C}_v = 4.768 \times .0136 = .0648 \\
\hline
\multicolumn{2}{l|}{\Sigma(V) = 6.732 \text{ cu.ft.}, \quad \Sigma(\mathbf{C}_p) = .1375} & \Sigma(V) = 6.732 \text{ cu.ft.}, \quad \Sigma(\mathbf{C}_v) = .1005 \\
\multicolumn{2}{l|}{\mathbf{C}_p = \dfrac{.1375}{6.732} = .0204} & \mathbf{C}_v = \dfrac{.1005}{6.732} = .0149 \\
\multicolumn{2}{l|}{\mathbf{C}_p = \dfrac{\mathbf{C}_p}{\gamma} = \dfrac{.020}{.0732} = .02787} & \mathbf{C}_v = \dfrac{.0149}{.0732} = .2035
\end{array}$$

$$x = \frac{.0204}{.0149} = \frac{.2787}{.2035} = 1.37.$$

The decrease of the gas constant R to the value R_r is proportional to the contraction during combustion. The maximum difference in this case is about 4% of R. The specific heat has increased slightly, while x is slightly lower than for the fuel mixtures, but the difference is negligible in view of the uncertainty existing with regard to the variation of specific heat with pressure and temperature. *There is, therefore, no considerable error involved in the assumption made in some of the computations of Part I, that the values of* R, c_p, c_v *and* x *are the same before and after combustion.*[1]

II. Older Views Concerning Combustion in Gas Engines

The peculiar views concerning the process of combustion in the gas engine, laid down by Otto in his original patents, have from the beginning led to very divergent opinions, which latter were on the one hand based in part upon theoretical considerations, and in part also upon the results of experiments on actual engines. The question, at first of interest merely from a scientific standpoint, became of the greatest practical importance at the instant it became apparent that the validity of the patents which covered Otto's claims depended entirely upon the possibility of carrying out the combustion process as specified. Both the students of thermodynamics and the engine builder at that time took up the study of the internal energy phenomena connected with combustion in the gas engine with the greatest thoroughness, and although most of the data obtained related to the older models of engines, many points of general interest and lasting value were brought to light.

In the following paragraphs a brief resumé of the final results of the most important investigations concerning this subject is given. The differences of opinion therein expressed deserve special attention, particularly in the cases in which they concern thermal phenomena obscure or incompletely investigated even to-day.

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[1] Corroboration of this will be found in Zeuner, Thermodynamik, I, p. 401.

1. Stratification of Charge and Retarded Combustion.

Otto, and Patent No. 532: "By first introducing air and afterward the charge, such stratification of the charge results that the latter near the cylinder head is composed of pure, readily combustible mixture, while near the piston it consists practically of burned gas. This is done for the purpose of retarding the combustion and to decrease maximum pressure and temperature. The ignition flame ignites only the particles of charge in its immediate proximity; after that the flame is propagated to the next charge particles, but its progress is the slower the further these combustible particles are apart, that is, the nearer the flame approaches the piston face. The burning particles of the charge transmit the heat evolved to the envelope of air surrounding them. The resulting tendency to expansion produces an increase of pressure which drives the piston. Since this pressure is a result of successive combustion of individual charge particles, it is produced gradually. Its action therefore is not similar to that of pressure suddenly produced by explosion of a charge and it is consequently also not accompanied by the unavoidable shocks and heat losses found in the 'explosion' engine."

Slaby thoroughly agreed with Otto's views regarding stratification and "slow" combustion of charge:[1]

"I am confirmed in this opinion by two arguments, first, the nature of the explosion line and second, the height of maximum pressure immediately after ignition. Careful investigation of indicator diagrams from Otto engines has shown that the closest approximations to the expansion curves are on the average obtained by use of the (polytropic) exponent $n=1.3$. It follows directly from this, however, that the equation of state is such as to indicate continually a greater influx than efflux of heat. If, on the other hand, the wavy expansion lines of the diagrams obtained from Hugon and Lenoir engines are replaced by average expansion lines, close approximations will show for the Lenoir engines $n=1.4$, and for the Hugon $n=1.6$. Consequently no influx of heat has taken place during the working stroke, and the stock of heat existing in the gas at the moment of maximum pressure is only partly converted into work, while a second and large part is transmitted to the cooling water. In both the Lenoir and Hugon machines, however, we have a single explosion of the entire charge followed by expansion. The expansion lines of the Otto diagrams on the contrary under all conditions point strongly to an influx of heat, and to explain this fact by assuming gradual combustion of the peculiarly arranged charge seems to me most plausible.

The assumption of gradual or slow combustion, however, finds further support in another way. The maximum explosive pressures in Lenoir and Hugon engines approximate 4 atmospheres (60 lbs.). On the basis of the excellent investigations made by Tresca on the composition of the explosive mixtures used in these engines, I have computed the pressures that should result from complete explosion and have obtained values which differ but little from those actually observed. The computed pressures exceeded those actually obtained by from 1 to 1½ atmospheres. The same computation made for Otto engines shows theoretical pressures of 17 atmospheres (240 lbs.), while actual indicator diagrams only show from 9 to 10 atmospheres (135–150 lbs.). (Slaby in another investigation has shown that of the total heat available in the charge, 56% are evolved during explosion and 44% during expansion in an Otto engine, while the corresponding figures for the Lenoir engine are 65 and 35% respectively.) This difference is perhaps indubitable proof that the first ignition does not at once produce an explosion of the total enclosed charge."

To further clear this question, Professor Slaby for nearly ten years carried on a series of scientific tests, on an 8 H.P. Deutz engine of the older model, to determine the heat distribution in the cylinder. The results of this highly important investigation were published by Slaby in his "Calorimetrische Untersuchungen über den Kreisprocess der Gas Maschine,"[2] from which the writer abstracts the closing sentences of Part IX (p. 161, Period of Ignition), in order to show how far Slaby's own original views were modified by his later work.

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[1] Verhandlungen des Ver. zur Beförderung des Gewerbefl., 1879, p. 38.

[2] Leonhard Simion, Berlin, 1894.

> "Based upon the results found, the actions during the ignition period in the Otto engine may be described as follows:
>
> 1. The ignition flame which is admitted to the cylinder at the inner dead center position of the piston first causes the combustion of that readily ignitable part of the total charge located in the ignition port. Indicator diagrams taken with double springs free from friction show this action by a sudden increase of pressure, although the latter may be comparatively small in amount.
>
> 2. As a result of this initial partial explosion there is formed a strong flame which, with piercing effect, strikes out into the combustion chamber proper and fires the rest of the charge with considerable lower velocity of flame propagation.
>
> 3. The velocity of propagation of the explosively formed flame depends (*a*) upon the average gas content of the charge, increasing or decreasing with the former; (*b*) upon the piston speed of the engine, the mechanical agitation produced increasing the velocity of propagation; and (*c*) upon the location of the combustible parts of the charge, that is, upon the degree of uniformity of the charge.
>
> 4. Combustion is complete after the piston has passed over a small part of its stroke, the time interval being from .03 to .06 seconds. The termination of the combustion period coincides with the point of maximum mean temperature in the cycle. Diagrams free from friction, and for which the ratio air to gas is about 6, clearly indicate the beginning of the expansion period which is then completed without further influx of heat.
>
> 5. The maximum temperatures do not exceed 2900° F., and since dissociation of the gas concerned does not begin to take place except at considerably higher temperatures, it may be held that dissociation does not occur.
>
> 6. Even during the combustion period the flame comes in contact with the cylinder walls and a part of the heat that has become available is conducted away by them. But since this only amounts to from 8 to 10% of the total heat, we may conclude, in view of the considerable difference in conductivity of the metal on the one hand and of the gases of combustion on the other, that the contact between the two is neither thorough or very rapid. The combustion therefore very largely takes place in the inner core of the charge, surrounded by an envelop of indifferent gases, Otto's own theory concerning the combustion process in his gas engine, in which he held that stratification of the charge (non-homogeneous concentric arrangement) actually existed, and that this was mainly the cause of the excellent economic showing of the machine, therefore seems entirely correct. The question whether the non-uniformity of the charge consists in stratification from the cylinder head toward the piston, or from the center of the charge toward the side walls, affects the case but little and will hardly be definitely solved. The view, formerly also held by the writer among others, but never to his knowledge held by Otto himself, that 'after-burning' extended into the period of expansion must, however, be considered erroneous."

The Deutz Tests. To prove the correctness of Otto's views, a 4 H.P. Deutz engine was so arranged that the mixture could not only be exploded by the usual ignition flame in the cylinder head, but also through a port which had its opening within the cylinder close to the piston in its inner dead center position. In these tests the usual ignition from the cylinder head always gave normal diagram with explosion at the dead center, while with ignition from the side the diagrams always showed an irregular and retarded combustion, the engine horse-power decreasing by one-half. If the engine was allowed to draw in first the fuel mixture and at the end some air, ignition from the cylinder head failed completely, while lateral ignition acted as before. These actions seem to justify the conclusion that the original method of charging proposed by Otto actually results in a charge which grows leaner toward the piston.

Dewar. During the patent litigation in England concerning the 4-cycle monopoly, Prof. Dewar and several other experts tried to substantiate Otto's views by taking samples of the mixture from various parts of the cylinder and examining these with regard to their combustion and combustibility. The result, in general, showed that the mixture at the place of ignition next to the cylinder head showed on the average 10% of combustible gas, while near the piston an average of only 5% was found, and an intermediate location averaged 7%. The sample from the location first mentioned ignited very readily, while those from the intermediate location could be ignited only with difficulty, and the mixtures taken near the piston failed altogether.

Teichman somewhat later repeated these eudiometric tests on a still larger scale.[1] The samples were as before taken at the instant of maximum compression from three places of the combustion space and electrically ignited in a eudiometer pipette. Of the resulting products of combustion the water vapor will condense to water while the carbon dioxide, owing to contraction, occupies a smaller volume. The total contraction is a relative measure of the combustible gases contained in the original sample. Teichman, for example, finds for the mixture taken from the explosion port (rich mixture) a contraction of from 12–14%, for the samples taken half way (average mixture) from 6–8%, and for the mixture near the piston face (lean mixture), only from 2–3%. Other tests consisted in taking indicator cards with varying location of the igniter and several forms of explosion port, and, to judge from the shape of the card, both the effectiveness of the ignition and the behavior of the fuel mixture. The final results of these tests are stated by Teichman as follows:

> "I think I have proven that the various parts of a gas engine charge introduced one after another will not mix uniformly, but that the charge will show a composition varying with the locality. Although the composition is not at all definite and cannot in any given case be predicted with any degree of mathematical accuracy, we do know that with proper form of cylinder and a certain method of charging we may expect to find with certainty in any given locality a mixture which will meet certain stated requirements. I further find that the local differences in the composition of the charge are not destroyed by compression, that they have a marked influence upon the entire progress of the combustion, and finally that by certain methods of charging, certain forms of combustion chamber and of explosion ports, the combustion may not only be influenced, but to a certain extent suitably controlled."

2. For and Against Dissociation.

Wedding (in a discussion following the article by Slaby, mentioned p. 542): "It is difficult to understand how, after the strong shaking up that the entire charge must receive from the first explosion, there should still be local differences in composition sufficiently marked to cause the remainder of the combustion to proceed gradually, and also hard to see why the diffusion is not sufficiently perfect to assume the combustion complete at least after a second explosion. It seems to me that the law of combustion established by Bunsen, according to which only so much gas can burn as is required to maintain a certain temperature, at which temperature *dissociation* of the products of combustion sets in and further combination (combustion) cannot take place, offers a sufficient explanation for the facts cited by Prof. Slaby. That which Bunsen has proven in the case of carbon monoxide and air, certainly also applies to illuminating gas and air."

Clerk.[2] "The cause of the sustained pressure shown by the diagrams is not slow inflammation (or slow combustion as it has been called), but the dissociation of the products of combustion, and their gradual combination as the temperature falls, and combination becomes possible. This takes place in any gas engine, whether using a dilute mixture or not, whether or not compression is used before explosion, and indeed it takes place to a greater extent in a strong explosive mixture than in a weak one."

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[1] Z. d. V. D. I., 1887, p. 271.

[2] Proceedings Inst. Civil Engrs., 1881–82.

Slaby.[1] "It is well known that all gaseous combinations break up into their constituent parts at a certain temperature, and that at this temperature these constituents are not again capable of combination. There is no doubt that dissociation plays a certain part in the action of a gas engine. As a result of the explosion, a strong influx of heat raises the temperature to such a point that some of the parts of the gaseous charge will commence to break up; beyond this point combustion cannot proceed and a further increase of temperature is hence impossible. This process is not combined with a loss of heat, because as soon as the temperature in the engine falls, the dissociated particles may again unite and thus render further combustion possible. The result of dissociation, however, is that only part of the illuminating gas really burns at the moment of explosion, while a further part does not burn until after expansion has started. According to this, dissociation would mean that there is an influx of heat along the expansion line and not a withdrawal. But there are two agencies operating at the same time: the cooling water abstracts heat, while dissociation, or rather the disappearance of it, furnishes heat. As a final result, heat may either be supplied or abstracted. The nature of the expansion line will show whether one or the other is the case. . . ."

Schöttler.[2] "I have never been able to thoroughly believe in any considerable effect due to dissociation, for if combustion really ceases at a certain temperature, it must recommence when the temperature sinks below this level, and presupposing uniform composition of charge, the expansion line must therefore necessarily be an isothermal as long as combustion is not complete. That is, however, never the case in a gas engine. But above all, the temperatures at which, according to Mallard and Le Chatelier, dissociation ensues, are always much higher than the maximum temperatures which occur in a gas engine. The latter is probably always below 2700° F. and in no case much higher."

Witz.[3] "It is not necessary to fall back upon the phenomena of dissociation to explain the prolonged influence of the burning part of the charge upon the burned gases. As a matter of fact, the condition mentioned occurs under circumstances which render dissociation impossible, since the temperature in the cylinder does not exceed 2600° F."

Mallard and Le Chatelier. "The cooling curves obtained by us have enabled us to determine with certainty that when a quantity of carbon dioxide which was raised to combustion temperature, is allowed to cool in a closed vessel, the dissociation which occurred at the moment of combustion steadily decreases until the mean temperature of the gas is 1800° C. (3270° F.). That at least is what occurs when the density of the carbon dioxide corresponds to that at a pressure of 50 cm. Hg. at 15° C. (19.7″ Hg. at 59° F.). The average temperature of the gas at which dissociation disappears decreases with the density of the gas; it amounts to only 1600° C. (2920° F.) when the density corresponds to that at a pressure of 28 cm. (11″) Hg. at ordinary temperature, and is only 1160° C. (2120° F.) when the pressure is 13 cm. (5.2″) Hg. We have also determined the effect that admixtures of other gases have upon the dissociation of carbon dioxide. The action of carbon monoxide, oxygen, and nitrogen is the same as if the pressure of the carbon dioxide had been diminished, and the mean temperature at which dissociation disappears decreases, all other conditions being the same. This effect is more strongly marked with nitrogen than with the other gases, and is more pronounced with oxygen than with carbon monoxide. As far as water vapor is concerned, we have not been able to prove any considerable degree of dissociation. It may be possible that the latter is obscured by the rapidity of condensation, in any case it must be small in amount even at 6000° F., which corresponds to the combustion temperature of the true oxy-hydrogen mixture in a closed vessel."

3. Uniform Mixture and Rapid Combustion: for and against the use of the Explosion Port.

Otto, and Patent No. 2735. "When an easily inflammable mixture only contains just sufficient air for its complete combustion, the combustion in the mixture after ignition will be practically instantaneous. If a strongly explosive mixture of this kind is diluted by an addition of air, the mixture remains inflammable, but the combustion of the entire charge is no longer instantaneous, requiring a longer or a shorter time depending upon the amount of air added.

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[1] Lecture before the General Convention of Gas-men, 1883.

[2] Z. d. V. D. I., 1886, p. 253.

[3] Ann. de Chimie et de Physique, 1883, vol. 30.

The charge in a gas engine cylinder, not stratified but of more or less uniform composition, must therefore be considered a diluted mixture of this kind. The charge may yet be readily ignited, but the combustion proceeds at a slower rate than that found in a stratified charge, because the rapid propagation of the combustion due to the explosion of the rich mixture near the place of ignition no longer occurs. It is possible, however, to explode such a diluted uniform mixture with the required greater rapidity, when, instead of starting the combustion in a single small locality, ignition is made to take place at the same time over an extended distance or area. For this purpose the combustion space of the cylinder which contains the lean gas mixture is connected with a branch chamber (port) which contains a rich mixture. If the latter is now ignited, the flame produced by this explosion will shoot out of this branch chamber or port into the combustion chamber proper and in its path will induce ignition in the diluted charge with rapidity and in all directions. By varying the volume ratio of the rich mixture to the lean main charge it is possible to regulate the rapidity of the combustion of the entire charge, thus adapting the duration of the combustion to the piston speed or controlling the pressure at beginning of expansion."

Clerk. "It does not matter whether the mixture used is rich or weak in gas; the rich mixture can be fired slowly and the weak one rapidly, just as may be required. The rate of ignition of the strongest possible mixture is so slow that the time of attaining complete inflammation depends only on the amount of mechanical disturbance permitted.

Fig. 685, a diagram from an Otto engine, shows what happens when the ignition comes late and the movement of the piston overruns the rate of the speed of the flame. The normal lines are those in which the rise is almost straight up from the point of the beginning of the ignition; they are marked *a* and *b*; the line *c*, although commencing from the beginning of the stroke, does not record the maximum pressure till the piston has moved forward one-third of its stroke, while the line *d* does not depart from the compression line until one-third of the forward movement, and does not attain its maximum till near the end of the stroke. In the last case the ignition has been missed until the piston is in rapid motion, and consequently the flame is at first unable to overtake it. This slow combustion, or rather slow ignition, in the gas engine must be avoided, and every effort should be made to complete the combuston as soon as possible after ignition. The perfection of slow combustion would be attained when the flame spread just as rapidly as the piston moves forward, and the pressure was never raised above that due to compression. The pressure diagram would then give the ideal results of 'gradual expansion of gases' and a 'perfectly sustained pressure.' But this is just the condition of greatest loss of heat: sustained pressure means sustained, indeed, increasing temperature, and the object to be attained in a good gas engine is to produce the most rapid possible fall of temperature due to work performed, to keep the mean temperature as low as possible, and it is only so far as this is successfully done that economy is possible. Slow inflammation causes loss of heat and power; rapid inflammation reduces the loss to a minimum while attaining the maximum possible power.

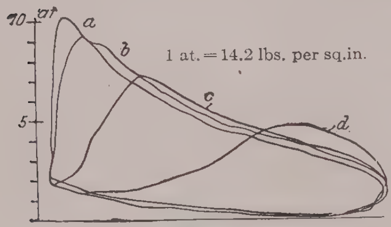

Fig. 685.

The modern gas engine does not use slow inflammation (or slow combustion, if the term be preferred), but when working as it is intended to do, completely inflames its gaseous mixture under compression at the beginning of the stroke. By complete inflammation is meant complete spread of the flame throughout the mass—not complete burning or combustion. If by some fault in the engine or ignition arrangement the inflammation is a gradual one, then the maximum pressure is attained at the wrong end of the cylinder, and great loss of power results."

Witz, in a series of experiments carried on in a free piston engine whose velocity could be changed at will, found that after-burning occurred for every composition of charge and manner of charging, but he also found that the more rapid the expansion of gases, i.e., the higher the piston velocity, the more rapid the combustion and the greater the maximum pressure.[1] From this he reaches the following conclusion:

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[1] Ann. de Chimie et de Physique, 1883.

"This is a law of great importance concerning the theory of gas engines. As a matter of fact, however, the marked influence of the piston speed is secondary to the effect produced by the walls, for if this were not the case how could rapidity of expansion affect explosion phenomena as indicated? This can only be due to cooling by contact with the metallic walls, which cooling, lasting a shorter or longer time, abstracts heat from the charge and reduces the violence of the reaction. The rapidity of combustion is, however, not the only thing affected. The influence of the cooling action also reduces the diagram area developed, the work done is decreased, and the efficiency drops, as was shown above. Therefore, in order to utilize the greatest possible portion of the heat contained in the explosion charge, it is important to expand the burned gases in the shortest possible time and the superficial area of the enclosing cylinder walls should be reduced as far as possible, i.e., the ratio $\frac{\text{superficial area}}{\text{volume of charge}}$ should be a minimum. . . . The influence of the cylinder walls is consequently the great regulator of combustion phenomena. It is capable of accelerating or retarding combustion. Any dilution of the charge naturally makes this effect more marked, because the mass of inert gas in which the explosive charge may be considered suspended acts, like the walls, as a cooling medium, but after-burning may also take place in an undiluted mixture. This conclusion, logically derived also from other tests, seem to us important; it both invalidates and supports the theory warmly advocated by Clerk. We agree with the eminent English engineer in that the combustion should not be purposely retarded; this retardation is an imperfection which should not be tolerated, and Otto is in the wrong when he advocates it. Unfortunately, however, this retardation, this so-called after-burning cannot always be completely avoided, because—as held by Clerk—the combination of the dissociated components of the charge only permits of a gradual development of the heat generated by the combustion, while the writer holds that the effect of the cylinder walls can only be toned down but never completely eliminated."

Ernst Körting, in conjunction with Professor Frese, made a number of tests on a vertical 4-cycle engine of his own make, during which the positions of igniter and of inlet (explosion) port were changed alternately between the cylinder head and the piston. Both the indicator cards obtained and the power capacity shown were equally satisfactory for all of the tests, which could only have been the case with a very uniform mixture. In connection with these tests, Körting says:[1]

"The best efficiency is attained when the distribution of the charge is such that the fuel is completely burned, and, further, such that the development of pressure is the most rapid possible. . . . Examining a stratified charge with reference to its combustion, we find:

1. That there is a very rapid combustion of the localized main part of the pure charge. This is necessarily accompanied by the development of a very high temperature combined with considerable local cooling and corresponding heat losses.

2. That, since combustion must necessarily stop in the zone of transition from one gas to another as soon as dilution in any one place has exceeded a certain limit, the use of a stratified charge must be accompanied by corresponding unavoidable losses due to incomplete combustion.

Another factor speaking strongly against the use of stratification, is the fact that the form of cylinder head called for to carry out this method of charging is with respect to cooling very bad.

The occurrence of after-burning, besides being partly due to the influence of the cylinder walls, may in small part also be explained on the basis of an observation made by Mallard and Le Chatelier, according to which, under conditions of gradual progress of ignition, the chemical process of combustion at any given point lasts longer than if the combustion had been explosive. Finally, there is the possibility that imperfect mixing may lead to the formation of several single 'tongues' of flame, such as are observed in the case of the combustion of illuminating gas in an unlimited air supply, and that this very fact causes after-burning, a thorough mixing taking place only during the combustion itself. In any case, no matter which of the three possibilities above outlined may be the cause of after-burning, the constructive means for remedying it is the same, and that is, to agitate the mixture as strongly as possible before combustion.

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[1] Z. d. V. D. I., 1886, p. 739, and 1887, p. 997.

. . . Whether the changing of the inlet port, in its capacity as explosion port, also has any effect upon the indicator diagram, remains problematical. It is known that the normal velocity of flame propagation, even in the most inflammable mixture, is about 6.5 ft./sec., while the pressure wave proceeds at the rate of 984 ft./sec. If, now, a mixture be ignited at the closed end of a port or passage, the combustion at the start causes a slight increase of pressure which drives the mixture ahead of it out into the combustion chamber with a velocity which is much in excess of that with which the ignition can possibly follow. Hence, in any case, whether the explosion port be long or short, the flame reaches the combustion chamber end without 'over pressure,' and will spread from here through the charge as though the ignition had originally ensued at the mouth of the port."

Schöttler bases his opinions on stratification of charge and retarded combustion upon the results of extended tests on engines of various makes and types, and does not deduce them entirely from his observation on the Otto engine. Besides the latter, he also examined Körting and Benz 2-cycle engines, both of which work with charge pumps and in which any considerable degree of stratification can hardly be expected. Professor Schöttler draws the following conclusions from his results:

"Otto's view regarding stratification is hardly tenable fundamentally. Taking into account the relative dimensions, the charge cannot very well be formed in any way except, as the piston moves outward, a current of air or of mixture flows into the cylinder, and this current, after passing the inlet port, must gradually expand, decreasing its velocity of flow. Simply considering that the ratio of inlet port to cylinder cross-section is as 1 to 40 in a 4 H.P. engine, one would be more apt to conclude that the conditions are much more favorable to thorough mixing than to the opposite. A stratification of charge would seem possible only when the cylinder is of such length that the incoming stream of charge is not able to overtake the piston. . . . It has, however, been shown in tests on other engines that the assumption of stratification of the charge is not at all necessary to explain the slow drop of the expansion line, irrespective of the question whether such stratification occurs in the Otto engine or not.

"To prove that stratification of charge does not exist, ignition in the Benz engine was produced in six different places in the cylinder. If the composition of the charge had been different in different places in any marked degree, this fact would have been shown by the diagrams. Nothing of the kind could be noticed, and the brake horse-power of the engine proved to be independent of the location of the igniter. In spite of the fact that not a trace of stratification appeared, the diagrams show the slow drop of the expansion line, indicating influx of heat during expansion and clearly pointing to strong after-burning. Entirely similar results were obtained with one of the older type of Körting engines in which not a trace of stratification as defined by Otto could be observed. It was noted, however, in the latter tests that the ignition, when produced immediately in front of the inlet port, was much more effective than that produced in any other place. This observation agrees with that made in the Deutz tests, where it was shown that ignition from the cylinder head was much better than the ignition from the side and nearer the piston. . . . Having now shown that after-burning is not the result of either dissociation or stratification of charge, the question remains as to why it occurs at all, or more correctly, why the expansion line falls much more rapidly in the older engines than in the new. This question has been answered by Witz on the basis of very careful tests. In his discussions upon this subject he comes to the conclusion that the shape of the expansion line is entirely controlled by the influence of the cylinder walls."

After a detailed consideration of the numerical results obtained by Witz in the tests above mentioned, Schöttler finally concludes:

'that the process of combustion in itself is the same in the newer as in the older engines. The success of the newer machines must therefore be sought in other directions, such as the use of compression, ignition at the dead center, which, in combination with compression, avoids shock, greater piston speeds, and finally the use of very lean mixtures, which has become possible by localizing a rich mixture in an ignition port, the action of which induces vigorous combustion."

III. Current Opinions Regarding the Process of Combustion

The controversy concerning "stratification" and "slow combustion" has long since died down. Even Otto himself, in his Patent No. 2735, had already abandoned a good share of his original views and long before the three first claims of his original Patent No. 532 were declared invalid, the charging and combustion process protected by these claims had become practically of no importance to the owners of the patent themselves. To-day the fundamental principle of gas-engine operation is to have

Pure and uniform mixture and most rapid combustion,

for upon this depends not only the efficiency of the entire combustion process, but also the specific engine capacity. Where it becomes necessary to dilute the charge in order to decrease the combustion temperature, or to increase the compression limit, atmospheric air only should be used for the purpose. Air in excess not only favors the combustion of the charge but also protects against over-heating those parts of the interior surface hard to reach by water-cooling, and finally increases the relative stock of heat, increasing the charge weight by keeping the temperature of the incoming mixture down.

The second most important rule of design calls for

Highest possible compression.

This, besides increasing the explosion pressure, also raises the mean effective pressure. On the other hand, it draws down the terminal pressure, decreasing the exhaust loss, see p. 13. Further, on account of the smaller volume of the charge when ready for ignition, the proportional admixture of burned gases to the charge and the loss of heat to the cooling water will both be less. Hence the general effect of high compression is to improve the thermal efficiency on all sides, and none of the means of improving the gas engine available to the designer has proven of greater effectiveness than the use of high compression.

In striving after high thermal efficiencies, the compression pressure has been increased step by step from 30 to 200 and even to 300 lbs., and the latter figure probably comes pretty close to what is practically possible. The heat of compression, on other counts so desirable, becomes a most serious drawback at the moment it raises the temperature sufficiently to reach the ignition temperature of the charge. The latter differs considerably for different fuels and for different fuel-air ratios of the same fuel, and hence in every case there is one definite most economical and yet safe compression limit. In general, the greater the content of carbon, and especially of hydrogen, in gases or gas-air mixtures, the more readily will they ignite. Consequently the vapors of the liquid fuels like kerosene and gasoline occupy the most unfavorable position in this respect in the list of fuels, and these are followed, in the order of allowable compression, by oil-gas, water-gas and producer-gas. The highest compression pressures are reached with mixtures the main combustible of which is carbon monoxide. This is particularly the case with blast-furnace gas, and this fact has materially helped in the economic development of the blast-furnace gas-engine, in which it was necessary to employ unusually high compression pressures from the beginning.

Low suction temperatures and effective cooling during compression are the only two means available to decrease the end temperature of compression, and the effective utilization of these, especially the latter, form a subject upon which designers will expend their efforts for some time to come. Neglecting the Bánki engine for the present, the cooling of the charge has up to the present been left entirely to the cylinder walls and the piston face. This is probably quite satisfactory even for high compressions in the smaller engines in which the superficial area is sufficiently large as compared with the volume of the charge. As the cylinders increase in size, however, the conditions grow less favorable (since the volume increases as the cube and the superficial area only as the square of the diameter), and a further complication results from the fact that the thicker walls of the larger cylinders offer a greater resistance to conduction, thus favoring local overheating. This is the reason why the smaller sizes of any type of engine can use a higher compression than the larger sizes, and also why the best economic efficiency of the smaller engines is not equalled in the larger machines. Resort is then necessarily had in such cases to special methods of cooling, the choice being either direct injection of water or artificial means of increasing the heat conducting surfaces of the combustion chamber, a scheme which was touched upon more in detail on p. 131.

As a third requirement for a perfect process of combustion may be mentioned

Proper form of combustion chamber and correct position of igniter.

The constructive means which may be used to meet this requirement are illustrated and discussed in Part II, pages 131 and 251, while p. 559 gives some figures based on tests.

The degree of imperfection of the combustion in gas engines may be numerically determined by accurately finding the content of combustible gases in the exhaust gases. An investigation of this kind is complicated and hard to carry through because the apparatus generally used in ordinary technical chemistry either gives results altogether too approximate or fails completely.[1] This is the reason why any definite information of general interest regarding the part that the heat loss due to incomplete combustion plays in the general heat balance, is only now and then obtained in tests very carefully prepared for the purpose. The figures available, however, admit of no doubt that in our present day engines the losses due to incomplete combustion are still very considerable, and certainly higher than is generally assumed. Table 142, compiled from tests made by Prof. E. Meyer,[2] may serve to prove this statement.

The remarkable thing in this table is the increasing heat loss with decreasing load. This, however, was in this case shown to be due to imperfect speed regulation (by means of taper cam, see p. 241, Part II). In general, the results show that, operating with illuminating gas at normal load (about 10 B.H.P.), the heat loss due to incomplete combustion amounts to from 3–5%, and at half load (about 6 B.H.P.) it is as high as 15%. The analysis of the exhaust gases made when operating on producer gas shows nowhere near as consistent results on account of the very dissimilar compositions of the fuel mixtures. At about 8 B.H.P. more than 13% of the heat in the fuel was shown in unburned particles in the exhaust gases; at about 7 B.H.P. one of the gas samples showed even 48% (supposing there is no error, as Meyer himself remarked). In general the heat losses due to this cause are greater when operating with producer gas than when working with illuminating gas. This is what may be expected in any case, both because of the varying gas-making conditions of the generator and the comparatively larger volume of fuel gas to be mixed with air each cycle. Improvement of mixing and ignition processes is therefore of special importance in the case of power-gas engines.

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[1] See Z. d. V. D. I., 1902, p. 948.

[2] Untersuchungen am Gas-motor, Z. d. V. D. I., 1902, p. 1303; also Mitteilungen über Forschungsarbeiten, 1903.

Ernst Körting at one time compared the various events of the cycle of a gas engine with those of the steam engine in order to facilitate forming an opinion of the former.[1] To the writer this comparison is so appropriate and generally valid even to-day, that a few of the points discussed will be inserted here.

> "Every engineer knows that the best efficiency in a steam engine is attained when the steam is admitted suddenly under the highest possible pressure, that is, when the diagram shows a vertical line in the dead center position of the piston, and when, after the required quantity of steam is admitted, it is suddenly cut off, so that any further supply of steam, i.e., heat supply, to the cylinder is completely cut off during the expansion period. All throttling of the steam during admission and all supply after cut-off simply decreases the efficiency.
>
> Every experienced engineer knows further that the high initial pressures, which in the steam engine occur at every dead center position very suddenly and with greater rapidity than in gas engines, are not, in a good engine, accompanied by jar or shock. The question of avoiding shock can, or rather need, never be considered as a reason for so controlling the process of combustion that the maximum pressure does not occur at the dead center position in gas

TABLE 142

HEAT LOSSES THROUGH INCOMPLETE COMBUSTION

KIND OF GAS.	ILLUMINATING GAS, HEATING VALUE = 567 TO 618 B.T.U. PER CU.FT.												PRESSURE PRODUCER GAS, HEATING VALUE = 121 TO 142 B.T.U. PER CU.FT.									
Compression.	Max., $\varepsilon = 4.98$ $p_c \sim 7.5$ at. = 107 lbs.				Mean, $\varepsilon = 4.59$ $p_c \sim 6.3$ at. = 90 lbs.				Min., $\varepsilon = 3.84$ $p_c \sim 5$ at. = 71 lbs.				Max., $\varepsilon = 4.98$ $p_c \sim 6.2$–6.9 at. ~ 88–98 lbs.			Mean, $\varepsilon = 4\ 59$ $p_c \sim 5.7$–6.2 at. ~ 81–88 lbs.			Min., $\varepsilon = 3.84$ $p_c \sim 4.7$–5 at. ~ 67–71 lbs.			
Load, B.H.P. . . .	10.24	10.11	8.15	6.13	9.77	9.60	8.12	6.09	10.36	8.30	8.26	6.19	9.04	8.17	8.06	8.81	8.17	6.08	8.17	7.12	6.16	6.11
Fuel ratio: $\frac{\text{air}}{\text{gas}}$ . .	7.89	8.30	10.51	11.43	8.74	8.48	10.31	12.08	8.88	10.96	8.44	11.11	1.05	1.12	1.25	1.23	1.19	1.32	1.22	1.19	1.56	1.53
Air in excess, % .	37	43	76	103	54	57	88	95	51.	61	56	74	−2	16	22	8	22	48	25	24	67	53
Heat loss due to incomplete combustion, %. . . .	3.2	3.2	15.4 (8.8)	15.1	2.8	5.9	(8.5)	15	4.0	1.9	4.7	14.4	4.5	5.3	12.9	2.9	13.1	8.5	1.8	48.3	7.8	11.3

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[1] Z. d. V. D. I., 1886, p. 757.

engines any more than in steam engines. The means employed to avoid shock under sudden pressure reversal in steam engines are well known, and it is sufficient to remark that exactly the same means are applicable to gas engines. In the latter, however, the solution of this problem is much simplified by the fact that the generation of pressure in the mixture is gradual and never very sudden. Accurate measurements on the rapidity of combustion in fuel mixtures of varying strengths and under varying degrees of compression are unfortunately lacking. In any case the velocity of flame propagation is so low that the diagram, Fig. 686, taken from the combustion of a charge that consisted of a rich mixture of illuminating gas and air, shows a combustion line distinctly inclined, in spite of the fact that the piston velocity near the dead center is low. The lines of this diagram clearly show the difference in the time interval required for the development of pressure, on the one hand between a highly explosive mixture, consisting only of illuminating gas and air, and, on the other hand, between a less inflammable mixture made up of two-thirds air and illuminating gas and one-third burned gas.

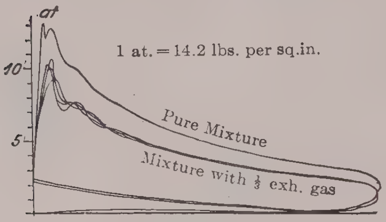

FIG. 686.

The application of the theory of construction of the steam engine to the gas engine therefore teaches us

1. That, in order to attain the best efficiency, the pressure development should be made as rapid as possible, the heat should all be developed in the dead center position, and during expansion no further heat should be supplied. So-called after-burning of the charge should, according to this, be avoided, and the writer would compare its effect to that of a leaky steam valve.

2. The charge should previously be highly compressed, because the useful effect of this increases in a greater ratio than the work required for compressing.

3. External cooling should be avoided.

The third rule can, unfortunately, not be fulfilled in gas engines. Instead of protecting the cylinders against loss of heat it becomes necessary for practical reasons to cool them and to cool them so strongly that the temperature difference between ignited charge and cylinder walls is considerably over 1800° F. The impossibility of avoiding cooling is the main reason for the principal differences in the construction of gas and of steam engines. A consideration of the circumstances thus controlling the design of gas engines brings out the following:

(*a*) While in a steam engine the highest possible steam temperature may be employed without unfavorably affecting the economy, a temperature increase in the gas engine is accompanied by heavy efficiency losses due to cooling. It therefore becomes necessary to keep the initial temperature of the fuel mixture down. Opposed to this is the fact that a low temperature is accompanied by low absolute efficiency, and this in turn means low economy. It should further be pointed out that a low temperature of the ignited charge can only be obtained through mixing with large quantities of inert gases, and that in a charge so diluted the pressure development must be so slow that the fulfilment of rule 1 above is out of question. The designer is called upon to steer a middle course between these too conflicting conditions, he should not go to the limit on either side.

(*b*) Since the amount of cooling is a function of time, it is one of the requirements for best efficiency of combustion to shorten the time during which cooling acts, i.e., the rotative speed of the engine should be as high as possible, a condition which is entirely without influence upon the efficiency of steam engines.[1] This requirement leads to the use of rich fuel mixtures in order to have the point of maximum pressure somewhere near the dead center position in spite of high piston speed.

(*c*) Since the amount of cooling is also a function of area exposed, a further requirement for best efficiency is to give such shape to the cylinder that the ratio of superficial area to volume enclosed shall be a minimum. This is again a point which is without influence upon steam-engine efficiency."

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[1] In the translator's opinion both this and the following point are intimately connected with the question of cylinder condensation in steam engines, and therefore have some effect upon steam-engine efficiencies.

Although the velocity of flame propagation in gas engine fuel mixtures differs considerably according to composition and other properties, and although little is known regarding this question, the fundamental relations, pointed out by Körting in (*b*) and (*c*) above, that exist between inflammability of charge, size of the combustion chamber, and rotative speed of engine, may to a certain extent be fixed mathematically.

Suppose that, with a certain velocity of flame propagation v, an ignition distance l (equal to the greatest length of the combustion chamber), and an engine speed of n revolutions per minute, the maximum combustion pressure occurs at the point a in Fig. 687, at which the combustion is complete. Then the time of combustion is

$$t=\frac{l}{v} \text{ seconds}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

and the angle passed through by the crank in this time is

$$\beta=tn\frac{360}{60}=n\frac{360}{60}\frac{l}{v} \text{ degrees.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

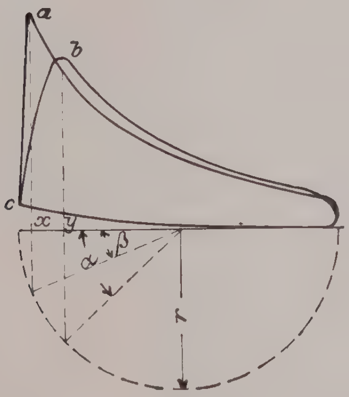

FIG. 687.

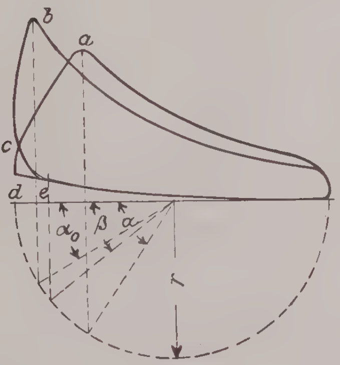

FIG. 688.

Assuming that the length of the connecting rod is infinite, the corresponding piston position will be

$$x=r(1-\cos\beta).$$

If now the rotative speed n be doubled, everything else remaining the same, the crank angle passed through to the point of complete combustion and the new piston position will be expressed by

$$\alpha=2n\frac{360}{60}\frac{l}{v} \text{ degrees}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

and

$$y=r(1-\cos\alpha). \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

The maximum explosion pressure then occurs at the point b and the ignition angle is hence doubled along with the rotative speed.

Area $\overline{abc}$ expresses the work lost on account of excessive piston speed. Equation (2) shows two ways in which this loss may be avoided, either by increasing v or by decreasing l. Compressing the charge works along both of these lines at the same time since it decreases the volume of the charge when ready for ignition (and consequently also l) and increases the inflammability of the compressed charge, (and hence also v). The higher the compression of the charge, therefore, the greater may be the piston speed or the leaner may the fuel mixture be made.

A further means of producing the maximum explosion pressure near the dead center, mostly used with the leaner gases and also in automobile engines, consists in advancing the point of ignition. If any given mixture, difficult of ignition, requires for its complete combustion a time interval equivalent to the crank angle α, Fig. 688, the maximum explosion pressure will evidently occur at the point a. If, however, instead of causing the ignition at the dead center, the charge is fired before the end of the stroke, while the crank is, say, in the position β, the combustion, proceeding with unaffected velocity, will then be complete after the crank has passed through the angle α_0 above center and hence the point of maximum pressure has been moved in from a to b, nearer the dead center. Advancing the ignition has, therefore, resulted in a saving of work equivalent to an area equal to [$\overline{abc} - \overline{cde} -$ the area above the $b-$ curve], without reference to any of the indirect effects of early ignition. This gain would only be wiped out when the areas of both diagrams become the same, but practical difficulties (shocks to the engine) put a stop to the extent to which ignition can be advanced. In any case this discussion shows that the proper time of ignition depends not only upon the properties of the charge, but also upon the size or shape of the combustion chamber and the crank velocity.

But it is not only the relatively low velocity of flame propagation which confines the allowable piston speed to within certain limits, there is also a second process, that of *diffusion* of the various gases making up the charge, which draws just as definite a limit. Every diffusion of one gas in another requires a certain length of time, and the time available for the process in the case of fuel mixtures is by no means as extended as is commonly assumed. The known diffusion coefficients (see p. 583, Appendix) are not of very great service in the solution of the question, because they throughout refer to kinds of gas as well as to pressure- and temperature-conditions not found in the cylinder of gas engines. But even these laboratory results show that the time of diffusion is of importance also to the gas-engine designer, and that he should take care at least not to cut the available time down to less than that of one stroke (by incorrect methods of governing, for example, see p. 241). The later investigations of Petreano, for instance, have shown that one volume of methane requires about six seconds to completely diffuse in one volume of air, while from ten to twelve seconds are required if the same volume of methane is injected into five volumes of air. The theoretical volume of air required for the combustion of methane is, however, twice the volume of air last mentioned. It is, of course, true that the heating up of the charge during the suction and charging strokes promotes diffusion, but the advantage is again partly counterbalanced by the fact that the increased pressure of the compression stroke has the opposite effect.[1]

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[1] The mobility of the gas molecules varies inversely as the density δ. For two gases having the same molecular weight and equal internal friction η, the velocity of diffusion is in general $D=\frac{\eta}{\delta}$. The same law is expressed in a different form by the equation of Clausius for the so-called "mean travel," $L=\frac{a^3}{\frac{4}{3}\pi s^2}$, in which a is the original mean distance between the molecules (that is, a function of δ) and s is the center distance between the two molecules at the moment of impact.

In any case it seems that even in slow-running engines ($n=150$ to 180) the time interval represented by two strokes is hardly sufficient, for complete diffusion, and that the latter extends into the expansion stroke. This fact alone would explain after-burning. In high speed engines this lack of uniformity in the mixing must of course be more marked, a condition which is proven by experience.

Considerations of this kind seem to make it desirable not to form the mixture in the working cylinder itself, but to previously mix the charge in a sufficiently large chamber unaffected by the movements of the piston and to draw the charge from this chamber as needed. This method has already been used for oil engines in several instances, but in gas engines it has found next to no application, probably mainly because of the added complication and the danger of explosion inherent in the use of such separate mixing chambers. That the employment of independent mixing chambers is of actual benefit in the process of combustion has been shown in some tests on a 4 H.P. Deutz engine made by Petreano with one of his mixing devices (see Fig. 396, p. 266).[1] Diagrams taken when the apparatus was connected in all cases show a more nearly vertical combustion line and a greater area developed than those taken during ordinary operation without previous mixing of charge. Combustion, completed at the inner dead center when the superficial area is least, is accompanied by a smaller consumption of cooling water, since the heat loss to the latter during expansion is less than it was before. To prove this Petreano also conducted tests without changing the cooling water in the jackets, that is, cooling by vaporization. After working for ten hours with 90% alcohol, the test being interrupted three times and distributed over three days, the amount of water that had to be replaced was .312 gallon, after operating with gasoline four hours without interruption, the quantity amounted to .65 gallon, while after $2\frac{3}{4}$ hours operation with 70% alcohol the temperature of the water had not gone beyond 176° F. The diagrams were from beginning to end uniformly good, but nothing is stated regarding horse-power, etc.

Forming the mixture outside of the cylinder in a special chamber is therefore a method which should receive some attention in practice, especially if, as seems probable, the rotative speeds of gas engines should continue to increase as they have done.

If it is not practicable to assist diffusion by extending the time allowed, it only remains to accelerate it by artificial means, that is, besides using a liberal excess of air, to employ some mechanical method of producing the same result. Excess air can only be used within narrow limits, the last scheme on the other hand deserves the special attention of the designer. The means at present adopted to mechanically aid in the diffusion of the gases making up a charge usually depend upon the principle of increasing the diffusion areas by minute subdivision of both kinds of gas. This method has been considered in greater detail in connection with the mechanical details used to carry it out, p. 195. Another scheme serving the same end, that of "agitating" the mixture by means of changing direction and cross-section of ports, stirring up the charge by sending a vigorous current of air or gas into the combustion chamber, etc., deserves to be much more widely tried than is at present the case.

In the Diesel constant-pressure engine, the fuel mixture is mechanically agitated even during ignition and combustion by forcing a fine stream of air, under a pressure of from 90 to 150 lbs. higher than that existing in the cylinder, into the charge and promoting the general distribution of this secondary air supply by appropriate form of injection nozzle, etc. In the explosion engine, in which the charge should be uniformly diffused at the moment of ignition, the mixture is artificially agitated to any extent only during the suction stroke, during the compression stroke the agitation amounts only to that incident to the operation. An exception to this rule is found in the case of certain 2-cycle engines in which the gas is forced into the cylinder while the air is being compressed. The fact that, on account of the short period available for the formation of the mixture in these engines, their charges do not show less, but rather better, uniformity than those of the older 4-cycle engines, points clearly to the beneficial action of agitation upon the constituents of the mixture.

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[1] Z. d. V. D. I., 1887, p. 170.

Körting clearly proved this in some tests made on one of his 4 H.P. vertical gas engines, taking indicator cards at widely varying engine speeds and determining from the course of the combustion line the time of explosion or the inflammability of the charge.[1] The ordinary indicator diagram does not admit of such determinations, because the events discussed occur at the moment of minimum speed of indicator drum. Körting therefore modified the drum by moving a strip of paper under the pencil at constant speed (in this case about 24.5 ins. per sec.) and thus obtained pressure-time diagrams, the curve showing the pressure changes as a direct function on time. Two of Körting's diagrams, reduced to three-fourths original size, are shown in Figs. 689 and 690, the former for the highest speed, $n=155$, and the latter for the lowest, $n=62$. These diagrams show that as far as this engine is concerned (inlet port located directly under the cylinder head) the absolute duration of combustion decreases as the engine speed increases. Thus for $n=62$, Fig. 690, the time of explosion was .045 sec., while for $n=155$, Fig. 689, this time had decreased to .02 sec. The time of combustion therefore varies in nearly the inverse ratio with the r.p.m. or piston speed, in fact it decreases at a little greater rate than the latter increases. Körting seeks the reason for this fact in the greater degree of mechanical agitation that the mixture receives at higher engine speed. The entering velocity of the charge for $n=62$ was about 62 ft./sec., while for $n=155$ it was about 156 ft./sec. This increased velocity probably not only increased relative velocities of the gas particles

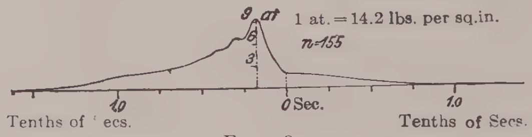

FIG. 689.

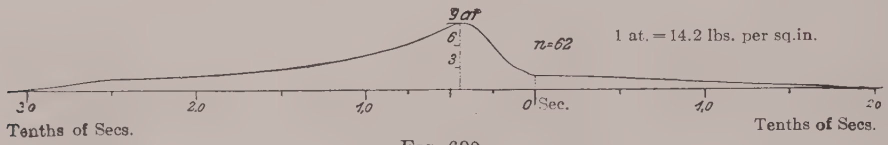

FIG. 690.

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[1] Z. d. V. D. I., 1888, p. 261.

among themselves in the inlet passage, but in the cylinder as well; and thus promoted diffusion and improved the uniformity of the charge.

It is quite possible, however, that the result shown may be partly due to the influence of other circumstances. For instance, higher piston speed is usually accompanied by higher exhaust temperature and hotter cylinder walls, and this may serve to pre-heat the charge and render it more inflammable. The weight of the burned gases remaining in the clearance decreases with the higher temperature, and since Körting kept the weight of the fresh charge the same at all speeds, it follows directly that the charge grew richer with every increase in the speed of engine. This in itself increases the inflammability of the mixture. It should be noted that in these tests, the speed variations were not obtained by exceeding the normal speed for which the engine was designed, but by starting it a speed below normal, and hence the volumetric efficiency of the suction stroke was nearly the same on all tests. The case is entirely different when with the increase of speed the normal speed of the engine is exceeded. In such cases the suction as well as the exhaust resistance losses increase, less fuel charge is drawn in, and more of the burned gases remain in the clearance space. This in turn decreases the charge weight and affects the purity of the mixture. Under such circumstances, instead of obtaining, as Körting did, decrease in the time interval required for combustion, we shall, of course, find considerable retardation.

A close examination of the pressure-time diagrams reveals the fact that they apparently controvert Körting's views regarding the "explosion port," see p. 547, while they confirm those of Otto. The full size diagram Fig. 691 ($n=62$) shows near the beginning of the explosion line that the original curve *ab* has a temporary break at the point *c*; after this the line shows the full rise to *d*. This indicates that the main explosion was preceded by a partial explosion, and the events were probably as follows: *a* to *b*, explosion in the explosion port, *b* to *c*, piercing of the main charge by the resulting needle flame. It takes a certain time for the flame to spread and for the effect to become apparent, hence the only slight increase of pressure from *b* to *c*. After the main charge is ignited at *c* the pressure rises very rapidly. This break in the line occurred, in all of the eight diagrams given by Körting, although the indication is not as strong at the higher speed, presumably because the flame spreads more rapidly under the changed conditions.

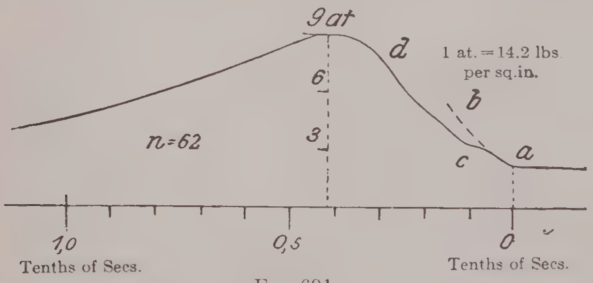

Fig. 691.

Very little reliable data is available regarding the *velocity of flame propagation* in gas-engine fuel mixtures. Many attendant circumstances, such as nature of the fuel, purity, composition, temperature and pressure of charge, shape and size of combustion chamber, kind and location of igniter, etc., have a marked effect upon combustion phenomena, and render the experimental results that have been obtained on this point so uncertain that they are of little value to the designer. In order to obtain values at all practical, some experimenters have used apparatus which reproduced more nearly the conditions existing in practice, and especially in place of the usual glass-tube models used in laboratories, have employed as combustion chambers metallic vessels approaching much more nearly to real working cylinders. In this way at least a few reliable comparative figures have been obtained.

Clerk carried on tests of this nature very soon after the appearance of the new Otto engine.[1] He used for this purpose a cylinder, without piston, 7 ins. in diameter and 8¼ ins. long. An igniter was screwed through the bottom cover, while connection with an indicator was made through the top cover. The charge fired consisted of pure fuel-air mixtures, and the pressure increase was recorded on the drum of the indicator which was revolved at constant speed. The curves thus obtained were hence similar to Körting's pressure-time curves and allow of the interpretation of the various events on the basis of time. The bundle of diagrams shown in Fig. 692 for instance was obtained with mixtures of air and Oldham coal gas. At the moment of ignition the charges were under atmospheric pressure, the temperature being from 60 to 66° F. It will be noted in conjunction with the table accompanying Fig. 692, that the mixture in ratio 1:14 gave the slowest, while 1:6 gave the most rapid combustion. Mixtures in which the ratio 1:6 was exceeded, that is, in mixtures still richer, the inflammability again decreased, as shown by curve *i*. From diagrams *g* and *h* we find a mean time of explosion of .05 sec. Since the height of the cylinder was 8¼ ins., the maximum velocity of flame propagation was accordingly about 13.8 ft./sec. Pure hydrogen-air mixtures in the ratio 2:5 showed velocities nearly three times this figure and developed explosion pressures exceeding 120 lbs. per sq.in. Clerk also computes the actual maximum combustion temperature from diagrams *g* and *h* as equal to 3256° F., and the theoretical explosion temperature, assuming that the heat is completely developed, as equal to 6870° F.

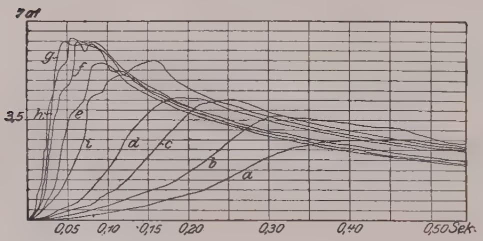

Diagram:	Fuel Mixture Ratio Gas to Air=
a	1 to 14
b	1 to 13
c	1 to 12
d	1 to 11
e	1 to 9
f	1 to 7
g	1 to 6
h	1 to 5
i	1 to 4

FIG. 692.

Soon after Clerk, Körting made similar tests with compressed air-illuminating gas mixtures.[2] The test cylinder used had a diameter of 6 ins., while the length was 5 ins. It was provided in the bottom, with an "explosion port" about 4 ins. long. The tests were made with varying degrees of compression and diagrams obtained, as before explained. Figs. 693 and 694 show the explosion lines of a number of them, together with the beginning of the cooling curve. The first row of diagrams was obtained with the richest mixture (1:5.42). Increasing the compression pressure from atmosphere to 30 lbs. apparently increases the time of explosion from .01 to .0125 sec. This retardation of the rapidity of explosion grows less marked as the mixture grows leaner. Thus for a ratio of 1:7.5, the diagrams of the second row show that while the time of explosion under atmospheric pressure (diagram *a*) is .032 sec., the time has increased only to .036 sec. (diagram *c*), when the mixture is originally compressed to 37 lbs. If, as in the case of Clerk's tests, the velocity of flame propagation is computed from the dimensions of the test cylinder and explosion port, we obtain the following results for the richest and the leanest mixtures:

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[1] See Clerk, "The Gas Engine," ed. 1886.

[2] Z. d. V. D. I., 1886, p. 875.

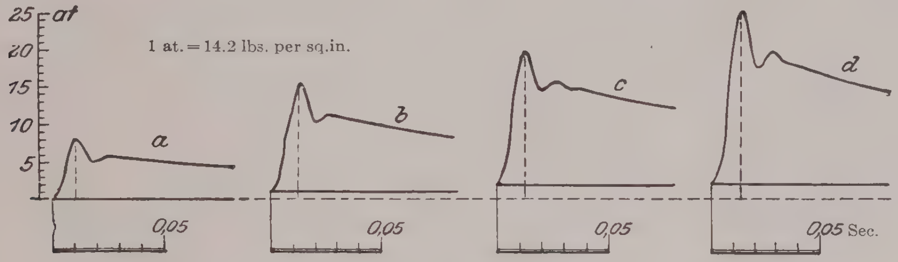

Fig. 693.

Rich mixture	Without compression	74.0 ft/sec.
	With compression	59.3 "
Lean mixture	Without compression	23.0 "
	With compression	20.5 "

The retarding influence of compression therefore seems to gradually decrease or to entirely disappear, as the mixtures grow leaner. That the kind of fuel used plays an important part in this phenomenon has, however, already been pointed out, pp. 536 and 557.

In order to study the influence that the *form of the combustion chamber* may have upon the nature of the combustion, Körting also made ignition tests in a tube 2.04 ins. in diameter and 16.4 ft. long. These tests showed that the combustions in the pipe are very irregular, being in some cases explosive and accompanied by violent noise, while in other cases they were slow and uncertain. It was also shown that the charge could be completely ignited in much shorter time when the ignition started from the middle of the length of the pipe than if the charge were ignited from one

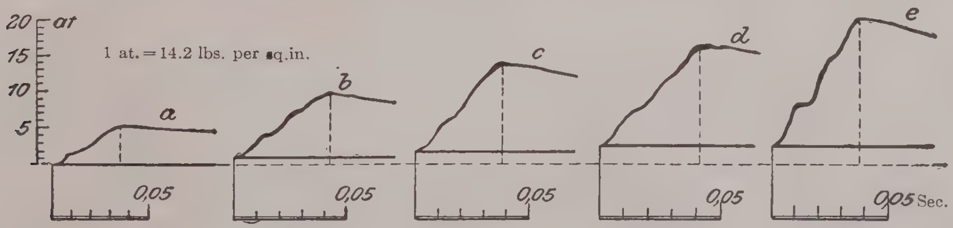

Fig. 694.

end. In all cases, however, the development of maximum pressure took a longer time in the long tube than in the short cylinder. Both the shape of the chamber and the location of the igniter therefore are of importance in the process of combustion. Theoretically, of course, the spherical form of chamber with the igniter at the center is the most desirable, because in this case the distances that the flame must travel to complete the ignition are of the same length in all directions and at the same time the shortest possible. In actual practice of course this ideal form of chamber can never be carried out, the next best shapes are the hemispheres or the short cylinder. In any case the igniter should be placed in the axis or center line of the chamber, as near as possible to the core of the compressed charge.

Another type of diagram which, besides the pressure-time diagram above mentioned, may serve to clearly bring out ignition and combustion events is the so-called "distorted" diagram Fig. 695, introduced by Prof. E. Meyer. To obtain such a diagram, the indicator reducing motion is offset with respect to the engine crank so as to bring the explosion line in the middle of the card, when the drum speed is the highest. With greater clearness even than the diagrams of Körting, the diagrams of Fig. 695 show the partial explosion preceding the main explosion, a phenomenon which is hard to understand except by the action of the explosion port in the cylinder head. Meyer himself explains its occurrence accordingly. The diagrams were taken from a Deutz horizontal kerosene engine.[1] On the basis of the data available, Professor Meyer found that the velocity of combustion of the kerosene vapor-air mixture was from 32 to 62 ft/sec, which is from 2 to 3 times the value found for lean illuminating gas mixtures. Another example of the distorted diagram, taken from a Haselwander engine, is given in Fig. 551.

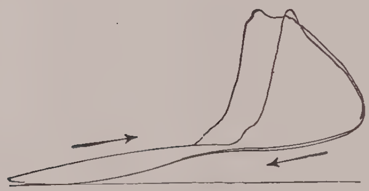

Fig. 695.

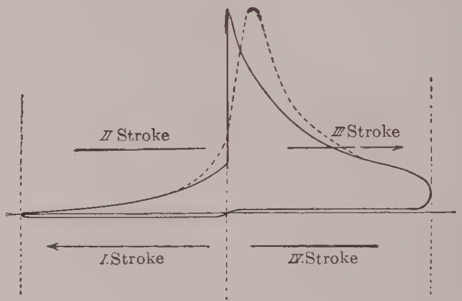

Fig. 696.

A further means of studying what happens during ignition and combustion is afforded by the "double-stroke" diagram which the writer has used since 1899 in connection with high speed engines, especially automobile machines. If the indicator is operated from the half-time shaft or from any other lay shaft running at one-half the speed of the crank-shaft, and if the reducing motion is so set as to bring the end of the compression and exhaust strokes in the middle of the travel, we will obtain an unusual diagram, the theoretical shape of which is shown in Fig. 696. In this diagram the pressure lines of the I and IV, and of the II and III strokes are recorded side by side. The combustion line is taken at the moment of maximum drum speed, as in the distorted diagram, and on that account shows any variation with great clearness. If the combustion were instantaneous (explosive) at the dead center, this line would rise vertically. On account of slow combustion, however, it inclines to the right, is as indicated in broken line in Fig. 696. The two double-stroke diagrams shown in Figs. 697 and 698 were obtained by the writer from a 3 H.P. gasoline engine running at 1400 r.p.m. The left side of the diagram records the compression stroke, the right side the explosion and exhaust strokes. Since, in such high speed engines, the spark is made to jump as early as one-half of the compression stroke, and since the speed of the indicator drum increases rapidly toward the center, the compression and combustion lines merge into each other. The compression curve alone (no ignition) is however clearly shown in Fig. 698, which records several miss strokes and late ignitions.[1] Diagrams of this kind can of course not be used to compute engine horse-power.

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[1] Z. d. V. D. I., 1895, p. 987.

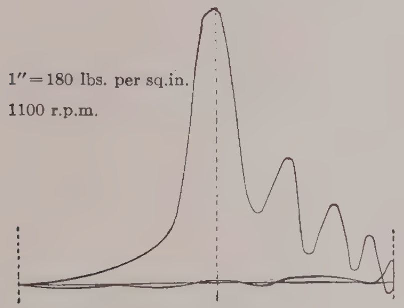

FIG. 697.

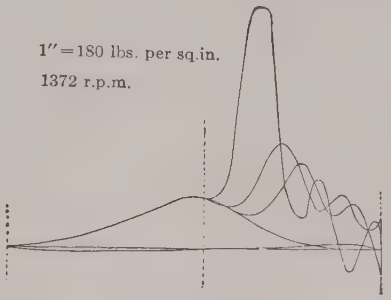

FIG. 698.

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[1] The writer's pamphlet, "Konstruction and Betriebsergebnisse von Fahrzeugmotoren," Berlin, J. Springer, contains a number of other diagrams of the kind together with a description of the indicator used

APPENDIX

A. THEORY[1]

I. Synopsis of Thermodynamics

1. Heat and Temperature and their Units. *Radiant heat* is considered as produced by transverse vibrations of ether and the latter are considered due to rapid vibrations of the molecules of the radiating body. Temperature is one of the several manifestations of heat which makes the existing degree of molecular movement (degree of heat) sensible externally. Heat and temperature are therefore different conceptions. As a measure of temperature we employ another manifestation of heat, that of the expansion of bodies. Of the latter, mercury and alcohol are the most used in ordinary temperature meters (thermometers). The subdivision of the distance between the two fixed points on the thermometric scale, freezing point and boiling point, is arbitrary. Besides the now quite extensively used scale of Celsius, which divides the distance into 100 equal parts, we further have the scale of Reaumur, which uses 80 equal divisions, and the Fahrenheit scale, almost exclusively used in this country and in England, in which the distance is divided into 180 equal parts. The latter is the oldest of the three. The conversion from one thermometric scale to another may be made by the use of the following equations:

$$C=\frac{5}{4}R=\frac{5}{9}[F-32^\circ]; \quad R=\frac{4}{5}C=\frac{4}{5}[F-32^\circ];$$

$$F=32^\circ+\frac{9}{5}C=32^\circ+\frac{9}{4}R.$$

The lower limit of commercial utility in the ordinary mercury thermometer is at about -40° C. $= -40^\circ$ F., the upper limit is usually not much above 250° C. = 482° F. When the space above the mercury is filled with nitrogen, brief readings up to 350° C. = 662° F. become possible, but the indication is uncertain above 280° C. = 536° F. Carbon dioxide thermometers may be used up to 550° C. = 1022° F. For the measurement of still higher temperatures, pyrometers are employed. Their action depends either upon longitudinal expansion or change of form of solid bodies, upon the expansion of air or upon the electro-motive force generated by thermo-elements. Another way of determining very high temperatures is to compute them from the specific heat of metallic bodies by calorimetric methods.

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[1] A brief and elementary guide for those readers of the book not thoroughly familiar with either Thermodynamics or Thermochemistry.

Suppose for instance that a wrought iron sphere of G pounds weight which had been exposed some time to the action of the exhaust gases just outside of the exhaust valve of a gas engine were quickly dropped into a non-radiating vessel containing W lbs. of water, then the temperature of the latter will increase from $t_1°$ to some value $t_2°$. If the mean specific heat of wrought iron (or any other metal) was c for the temperature range, $t_1°$ to $t_2°$, the temperature of the exhaust gases must have been

$$t = t_2 + \frac{(t_2 - t_1)W}{Gc} \text{ degrees.}$$

According to Weinhold, we have for wrought iron

$$c = .105907 + .00003269(t_1 + t_2) + .0000000110 8[t_1^2 + t_2^2 + (t_1 + t_2)^2].$$

Note that here t is in Centigrade degrees.

For temperatures less than −40° C. or F., alcohol thermometers, whose field of application ranges from −100° C. to +78° C. (−148° F. to +172.4° F.) are used.

Just as temperature is expressed in degrees, the quantity of heat is expressed in heat units (B.T.U., British thermal units). A heat unit is the quantity of heat required to raise the temperature of one pound of water under atmospheric pressure through 1° F. [In the metric system the heat unit (kilogram-calorie, or calorie for short) is the quantity of heat necessary to raise 1 kilogram of water through 1° C. The gram-calorie, sometimes called the small calorie, is equal to $\frac{1}{1000}$ of the kilogram-calorie.

2. Pressure, Density and Specific Weight or Gravity. The "heat carriers" in gas engines are gaseous bodies or mixtures of them, simply called "gas" for short. Every gas, owing to its tendency to expansion, exerts upon the walls of the vessel enclosing it, a pressure, the magnitude of which is expressed either by the height of a column of liquid maintained in equilibrium by it or by the pressure exerted upon unit area of the enveloping surface. The instruments used in the first system of measurement are called *manometers*, the liquid being mercury or, for very small pressures, water; those used in the second system are designated simply *pressure gauges*. The unit of pressure is the "atmosphere," in English units equal to 14.7 lbs. per sq.in., or, expressed in metric units, equal to 1.0333 kg. per sq.cm., at sea level. For technical purposes this value of 1.0333 kg. per cm. has throughout Germany been replaced by a unit equal to 1 kg. per sq.cm., equivalent to 14.2 lbs. per sq.in., which gives the conversion factor used throughout this book. The following table expresses the relation existing between the units when expressed in various ways.

True Atmosphere.				Metric Atmosphere.			
Kg. per sq.cm.	Lbs. per sq.in.	Inches, Mercury (Hg.)	Feet, Water.	Kg. per sq.cm.	Lbs. per sq.in.	Inches, Mercury (Hg.)	Feet, Water.
1.0333	14.70	29.92	33.91	1.00	14.20	28.74	32.75

A barometer reading of 29.92″ (=760 mm.) is usually taken as the *standard pressure*, while 32° F. (=0° C.) is taken as the *standard temperature*. The two together determine the *standard condition* of a gas (unless for certain purposes another standard set is chosen, see p. 6).

In practice the pressure of a gas is usually stated in pounds above atmospheric pressure, that is, it is really what might be called the "over-pressure" (gauge pressure). In thermodynamic investigations, on the other hand, the pressure is always used as *absolute pressure*, that is, measured above a vacuum.

Density (δ) of a gas is defined as the weight of a given volume of the gas divided by the weight of an equal volume of pure, dry air. The weight of 1 cu.ft. of a gas at 32° F. and 29.92″ Hg. (14.7 lbs. per sq.in.), is called the *specific weight* (γ). Since 1 cu.ft. of pure, dry air under the conditions stated weighs .08072 lb., we have that

$$\gamma = .08072\delta, \quad \text{. . . . .} \quad (1)$$

or

$$\delta = \frac{\gamma}{.08072}. \quad \text{. . . . .} \quad (2)$$

There are two other ways of expressing the value of γ. From p. 569 following, we have, eq. (19), that

$$\frac{Pv}{T} = R,$$

in which v = volume of 1 lb. of gas in cubic feet, P = pressure in lbs. per sq.ft., and T = absolute temperature in °F. = 461 + temperature as measured. R expresses the work done by 1 lb. of gas when heated through 1° F., the pressure P remaining constant. R is a constant for any one gas. We may write

$$v = \frac{TR}{P} \quad \text{or} \quad \frac{1}{v} = \frac{P}{TR},$$

but if v represents the volume in cubic feet of 1 lb. of gas, $\frac{1}{v}$ represents the weight of 1 cu.ft. of gas = γ, hence

$$\frac{1}{v} = \gamma = \frac{P}{TR}.$$

Now under standard conditions, P = 14.7 lbs. per sq.in. = 2117 lbs. per sq.ft., and $T = 461 + 32 = 493°$, therefore

$$\gamma = \frac{2117}{493\,R} = \frac{4.29}{R} \text{ lbs. per cu.ft.,} \quad \text{. . . . .} \quad (1a)$$

The second expression for the value of γ is derived directly from the law of Avogadro. According to this

$$\gamma = \frac{m}{358.17} \text{ lbs. per cu.ft.,} \quad \text{. . . . .} \quad (1b)$$

in which m is the molecular weight of the gas (see p. 580).

Under standard conditions (32° and 14.7 lbs.) v_0 cu.ft. of gas will weigh

$$G_0 = \gamma v_0 = .08072\delta v_0 \text{ lb.} \quad \text{. . . . .} \quad (3)$$

If however, a gas whose specific weight is γ, instead of being under normal conditions, is under a pressure of p pounds absolute or b inches of mercury absolute, at a temperature of $t°$, v cubic feet of the gas under the conditions given will weigh

$$G=\gamma v\frac{p}{14.7}\;\frac{493}{461+t}=\gamma v\frac{b}{29.92}\;\frac{493}{461+t}\text{ lbs.} \quad . \;.\;.\;.\;.\;.\;.\;.\;.\;. \quad (4)$$

The density of the gas under the same conditions will then be

$$\delta'=\delta\frac{p}{14.7}\;\frac{493}{461+t}=\delta\frac{b}{29.92}\;\frac{493}{461+t}. \quad .\;.\;.\;.\;.\;.\;.\;.\;.\;.\;. \quad (2a)$$

Under ordinary conditions, atmospheric air is considerably lighter than dry air on account of the water vapor carried.

3. Expansion, Absolute Temperature, and Specific Heat. If the temperature of a gas is raised from 32° to $t°$, while the pressure remains constant, the original volume v will expand to

$$v_1=v[1+\alpha(t-32)], \quad .\;.\;.\;.\;.\;.\;.\;.\;.\;.\;.\;. \quad (5)$$

or, if the temperature at the beginning is $t'°$, instead of 32°,

$$v_1=v\frac{1+\alpha(t-32)}{1+\alpha(t'-32)}, \quad .\;.\;.\;.\;.\;.\;.\;.\;.\;.\;.\;. \quad (5a)$$

α is called the *coefficient of expansion* at constant pressure. For changes at constant volume, the pressure changes according to a similar law. The *pressure coefficient* is slightly greater than the volume coefficient α, although the difference is practically negligible. For all perfect gases the value of α is nearly the same, that is

$$\alpha=\frac{1}{493}=.00203.$$

A volume of gas v_1 at a temperature $t°$ and pressure b inches of mercury, will therefore, when reduced to standard conditions, occupy a volume equal to

$$v_0=v_1\frac{493b}{29.92[493+(t-32)]}=\frac{v_1b}{29.92[1+.00203(t-32)]}. \quad .\;.\;.\;.\;. \quad (6)$$

This equation is valid whether heat is supplied or withdrawn. Assuming now that it were possible to cool a gas down so far as to make its volume zero, eq. (6) shows that the temperature would have to be decreased 493° below 32°, the freezing point. This point, −461° below the zero of the Fahrenheit scale, is called the *absolute zero*, and *absolute temperatures* are measured from it as a starting point. All computations in thermodynamics are based upon absolute pressure and absolute temperatures.

The *specific heat* (capacity for heat) is the quantity of heat (in B.T.U.) which must be supplied to one pound of a given body to raise its temperature 1° F. In the case of solid bodies, the specific heat varies with the temperature; in gases it is also dependent upon the pressure. A given body of G pounds weight requires a quantity of heat equal to

$$Q=G\int_{t_1}^{t_2}c\,dt\text{ B.T.U.}, \quad .\;.\;.\;.\;.\;.\;.\;.\;.\;.\;.\;. \quad (7)$$

to raise its temperature from t_1 to t_2, through a range of $t°$, the value of the specific heat c not being constant between the temperature limits given. In general, however, the variation in the value of c is very small and in many cases yet unknown, hence it is usual to assume a mean value of c, in which case eq. (7) reduces to

$$Q=Gct \text{ B.T.U.} \qquad (7a)$$

If G pounds of one fluid, having a temperature t and a specific heat c, is mixed with G' pounds of another liquid, having a temperature t' and a specific heat c', we may write the resulting temperature t_m, according to Richmann's rule, equal to

$$t_m=\frac{Gct+G'c't'}{Gc+G'c'} \quad \text{or, in general,} \quad t_m=\frac{\Sigma(Gct)}{\Sigma(Gc)}. \qquad (8)$$

The specific heat of gases at constant volume (c_v) is always considerably smaller than that at constant pressure (c_p) and we may write for all perfect gases

$$\frac{c_p}{c_v}=x>1; \quad c_p=\frac{2x}{m(x-1)}; \quad c_v=\frac{2}{m(x-1)}; \qquad (9)$$

$$c_p-c_v=\frac{2}{m}=AR. \qquad (10)$$

The conversion of the specific heats c_p and c_v rom the weight (pound) basis to the volume (cubic feet) basis is made by means of multiplying by the specific weight $\gamma=\frac{m}{358.17}$ (see p. 581) so that

$$\mathbf{c}_p=\gamma c_p, \quad \text{and} \quad \mathbf{c}_v=\gamma c_v.$$

TABLE 143

SPECIFIC HEATS AND CONSTANT R FOR PERFECT GASES

NAME OF GAS.	SPECIFIC HEAT.				$x=\frac{c_p}{c_v}$	R
	Per Pound.		Per Cu.ft. at 32° F. and 29.92″ Hg.			
	c_p	c_v	$\mathbf{c}_p$	$\mathbf{c}_v$		
Hydrogen ... H_2	3.43	2.43	.0191	.0136	1.41	770.2
Oxygen ... O_2	.217	.153				48.30
Nitrogen ... N_2	.245	.174				55.08
Carbon monoxide ... CO	.245	.174				55.21
Nitric oxide ... NO	.23	.163				52.47
Air, (dry and pure) ...	.238	.169	....	....	....	53.34
Water vapor ... H_2O	.48	.37	.0241	.0185	1.300	85.87
Carbon dioxide ... CO_2	.20	.155	.0245	.0189	1.293	35.13
Acetylene ... C_2H_2	[.346]	[.27]	[.0251]	[.0196]	[1.281]	59.42
Methane ... CH_4	.593	.468	.0265	.0208	1.270	96.47
Ethylene ... C_2H_4	.40	.33	.0354	.0292	1.210	55.19
Alcohol ... C_2H_6O	.453	.400	.0576	.0508	1.113	33.56
Benzol ... C_6H_6	.33	.305	.0713	.0659	1.082	19.80

Table 143 contains the specific heats for all of the so-called perfect gases likely to be used in the gas engine industry. Mixtures of gases having the same number of atoms have the same volume specific heats as the constituent gases; thus for a mixture of di-atomic gases, for instance, we have, from Table 143,

$$\mathbf{c}_p = .0191 \quad \text{and} \quad \mathbf{c}_v = .0136.$$

If a mixture consists of G_1, G_2, G_3, . . . , parts by weight or of v_1, v_2, v_3, . . . , parts by volume of various gases, so that $\Sigma(G_n) = 1$, or $\Sigma(v_n) = 1$, the respective specific heats of the mixture will be

$$c_p = \Sigma(Gc_p)_n, \quad c_v = \Sigma(Gc_v)_n; \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad (11)$$

$$\mathbf{c}_p = \Sigma(v\mathbf{c}_p)_n, \quad \mathbf{c}_v = \Sigma(v\mathbf{c}_v)_n. \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad (11a)$$

4. The Relations between Volume, Pressure, and Temperature. The Equations of State or Condition. For a certain given weight of a gas, the product of the volume v by the pressure p, or the product of the volume v by the density δ, is a constant as long as the temperature remains constant, that is

$$vp = v_1 p_1 \quad \text{(Law of Boyle)} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad (12)$$

and

$$v\delta = v_1 \delta_1 = 1.0. \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad (12a)$$

Hence

$$\frac{v}{v_1} = \frac{p_1}{p} = \frac{\delta_1}{\delta}. \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad (13)$$

As stated, eq. (13) only holds good as long as the changes are "isothermal," i.e., as long as the temperature remains constant. If, on the other hand, the temperature during the change increases from T to T_1, we will have

$$\frac{v}{v_1} = \frac{T}{T_1}, \quad \text{or} \quad \frac{p}{p_1} = \frac{T}{T_1}, \quad \text{or} \quad \frac{\delta}{\delta_1} = \frac{T}{T_1}, \quad . \; . \; . \; . \; . \; . \; . \; . \quad (14)$$

according to which the $\left\{\begin{array}{l}\text{volumes}\\ \text{pressures (densities)}\end{array}\right\}$ at constant $\left\{\begin{array}{l}\text{pressures (densities)}\\ \text{volumes}\end{array}\right\}$ are to each other as the absolute temperatures. (Law of Gay-Lussac.)

The first two equations of (14) can also be derived in another way. Thus from eq. (5) we have, if a volume v is heated from 32° to some temperature t at constant pressure, that the final volume will be

$$v_1 = v[1 + \alpha(t - 32)] = v[1 + .00203(t - 32)] = v\left(1 + \frac{t - 32}{493}\right). \quad . \; . \; . \; . \; . \quad (15)$$

Similarly, if during the same range of temperature the volume remains constant, we will have a final pressure equal to

$$p_1 = p[1 + \alpha(t - 32)] = p[1 + .00203(t - 32)] = p\left(1 + \frac{t - 32}{493}\right). \quad . \; . \; . \; . \; . \quad (16)$$

Substituting in eqs. (15) and (16),

$$T=493, \quad \text{and} \quad T_1=[493+(t-32)]=461+t,$$

we have as before

$$\frac{v_1}{v}=\frac{T_1}{T}, \quad \text{and} \quad \frac{p_1}{p}=\frac{T_1}{T},$$

or

$$v_1=v\frac{T_1}{T} \quad \text{and} \quad p_1=p\frac{T_1}{T}. \quad \ldots \ldots \ldots \quad (17)$$

If the temperature at the beginning of the change had been some temperature t' instead of 32°, v_1 (and by analogy also p_1) would have been directly found from eq. (5*a*).

Suppose that one pound of a gas, of volume v, is heated through 1° at a constant pressure of P pounds per sq.ft. If the final volume is v_1, and we designate the work done by the pound of gas in expanding from v to v_1, against the pressure P by R, we shall have

$$P(v_1-v)=R,$$

or

$$Pv\left(\frac{v_1}{v}-1\right)=R.$$

Now from (17)

$$\frac{v_1}{v}=\frac{T_1}{T},$$

hence

$$Pv\left(\frac{T_1}{T}-1\right)=Pv\left(\frac{T_1-T}{T}\right)=R.$$

But since we are heating only through 1°, $T_1-T=1$, hence finally

$$Pv=RT, \quad \ldots \ldots \ldots \quad (18)$$

R is a constant within ordinary limits for any one gas, but it should be noted that very low temperatures or very high pressures cause a variation in R, and (18) is then no longer strictly applicable. Eq. (18) is a combination of the law of Boyle with that of Gay-Lussac and may also be written

$$\frac{Pv}{T}=\frac{P_1v_1}{T_1}=R=\text{constant}, \quad \ldots \ldots \ldots \quad (19)$$

or

$$\frac{v}{v_1}=\frac{P_1T}{PT_1}. \quad \ldots \ldots \ldots \quad (19a)$$

For values of R, see Tables 143 and 147.

If a gas is compressed in a non-conducting cylinder, its temperature is increased from T at the beginning to some temperature T_1. This in itself, according to the law of Gay-Lussac, causes an increase in the pressure. The latter is hence increased in two ways, first, by the pressure on the piston and, second, by the effect of the heat of compression. Consequently the pressure exerted by the gas must increase faster than it would without the temperature rise. The reverse takes place when a volume of gas is expanded from v to v_1, in which case the work of expansion is done at the expense of the heat contained in the gas. This action decreases the temperature which decrease is accompanied by contraction, and hence the pressure falls faster than if the temperature during the expansion had remained constant.

According to the law of Poisson, we have for *adiabatic* changes of state, during which no heat is either supplied to or withdrawn from the gas, that

$$pv^x = p_1v_1^x; \quad . \; . \; . \quad (20) \qquad \frac{p}{p_1} = \left(\frac{v_1}{v}\right)^x, \quad . \; . \; . \quad (21) \quad \text{and} \quad \frac{v}{v_1} = \left(\frac{p_1}{p}\right)^{\frac{1}{x}}, \quad . \; . \quad (22)$$

in which p and v refer to the original state of the gas, p_1 and v_1, to the final state, and $x = C_p \div C_v$, the ratio of the specific heats of the gas in question.

These equations are best solved by means of logarithms, thus

$$\frac{p}{p_1} = \text{anti-log}\left(x \log \frac{v_1}{v}\right), \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad (21a)$$

and

$$\frac{v}{v_1} = \text{anti-log}\left(\frac{1}{x} \log \frac{p_1}{p}\right). \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad (22a)$$

By combining eqs. (20) to (22) with eq. (14) of Gay-Lussac, we may derive expressions giving the increase or decrease of absolute temperature with adiabatic increase or decrease of pressure. We have

$$\frac{T}{T_1} = \left(\frac{v_1}{v}\right)^{x-1}, \quad . \; . \; . \quad (23) \quad \text{and} \quad \frac{v}{v_1} = \left(\frac{T_1}{T}\right)^{\frac{1}{x-1}}, \quad . \; . \; . \quad (24)$$

or

$$\frac{T}{T_1} = \left(\frac{p}{p_1}\right)^{\frac{x-1}{x}}, \quad . \; . \; . \quad (25) \quad \text{and} \quad \frac{p}{p_1} = \left(\frac{T}{T_1}\right)^{\frac{x}{x-1}}. \quad . \; . \; . \quad (26)$$

If the original state of a gas is defined by p, v, and T, the end conditions, after an adiabatic change, are

$$p_1 = p\left(\frac{v}{v_1}\right)^x = \text{anti-log}\left[\log p + x\left(\log \frac{v}{v_1}\right)\right]; \quad . \; . \; . \; . \; . \; . \; . \; . \quad (27)$$

$$p_1 = p\left(\frac{T_1}{T}\right)^{\frac{x}{x-1}} = \text{anti-log}\left[\log p + \frac{x}{x-1}\left(\log \frac{T_1}{T}\right)\right]; \quad . \; . \; . \; . \; . \quad (27a)$$

$$v_1 = v\left(\frac{p}{p_1}\right)^{\frac{1}{x}} = \text{anti-log}\left[\log v + \frac{1}{x}\left(\log \frac{p}{p_1}\right)\right]; \quad . \; . \; . \; . \; . \; . \; . \; . \quad (28)$$

$$v_1 = v\left(\frac{T}{T_1}\right)^{\frac{1}{x-1}} = \text{anti-log}\left[\log v + \frac{1}{x-1}\left(\log \frac{T}{T_1}\right)\right]; \quad . \; . \; . \; . \; . \quad (28a)$$

$$T_1 = T\left(\frac{v}{v_1}\right)^{x-1} = \text{anti-log}\left[\log T + (x-1) \log \frac{v}{v_1}\right]; \quad . \; . \; . \; . \; . \; . \; . \quad (29)$$

$$T_1 = T\left(\frac{p_1}{p}\right)^{\frac{x-1}{x}} = \text{anti-log}\left[\log T + \frac{x-1}{x}\left(\log \frac{p_1}{p}\right)\right]. \quad . \; . \; . \; . \; . \; . \quad (29a)$$

The expression $\left(\log \frac{v}{v_1}\right)$, etc., may also be written $(\log v - \log v_1)$, etc.

TABLE 144

AVERAGE VALUES OF c_p, c_v, AND x FOR AIR AND PRODUCTS OF COMBUSTION

		c_p	c_v	$\frac{c_p}{c_v}=x$	$\frac{1}{x}$	$x-1$	$\frac{x}{x-1}$	$\frac{1}{x-1}$	$\frac{x-1}{x}$
Air, pure and dry		.2375	.1684	1.410	.709	.410	3.493	2.493	.291
For products of combustion from illuminating gas-air mixtures in the ratio of	1 to 6	.268	.198	1.356	.738	.356	3.809	2.809	.263
	1 to 9	.259	.189	1.370	.730	.370	3.703	2.703	.270
	1 to 12	.254	.184	1.380	.725	.380	3.632	2.632	.274

It is simpler in many cases to determine the end temperature T_1, from the general relation $Pv=RT$. It should be remembered in this connection that P represents absolute pressure in pounds per sq.ft., and v is the specific volume in cu.ft. of one pound of gas. If instead of v the actual volume V of the gas is given in cubic feet, so that the weight is G pounds, we will have

$$v=\frac{V}{G}=\frac{1}{\gamma},$$

and from $VP=GRT$, we then get

$$T_1=\frac{P_1V_1}{GR}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (30)$$

5. Examination and Construction of Pressure-Volume Curves. The theoretical pressure-volume curves are of importance in thermodynamic investigations because they indicate what the pressure and volume changes should be in theory, and consequently form a basis upon which to judge the real changes occurring in practice. To construct or draw the curves, the rectangular system of coördinates, *ox*, *oy*, Fig. 699, is employed, the x-axis (abscissas) representing a scale of volumes, and the y-axis (ordinates) a scale of pressures. These scales may be chosen to any convenient length per unit. From *o*, the origin, lay off a distance $v=oa$ equal to the original volume, and at *a* erect a perpendicular of length *ab*, so that $ab=p$ is the original pressure. The point *b* then represents the original state, designated by p, v, T, of a certain quantity, say one pound of the gas. If now we allow the gas to expand to say 5 times its original volume, the point *c*, at a distance equal to $5v=5ao$ from the origin along the x-axis, will represent the final volume v_1. The length of the ordinate through *c*, however, depends upon the equation of condition or state according to which the gas expanded. If, for instance, by means of a proper supply of heat, the pressure had been maintained constant, we will have $cd=ab$, so that $p_1=p$, and the curve would have been a straight line *bd* parallel to *ac*. This is known as a *constant pressure line* or an *isobar*. The heat supplied has of course raised the temperature from T to T_1, and the final state of the gas at the point *d* is given by p, v_1, T_1.

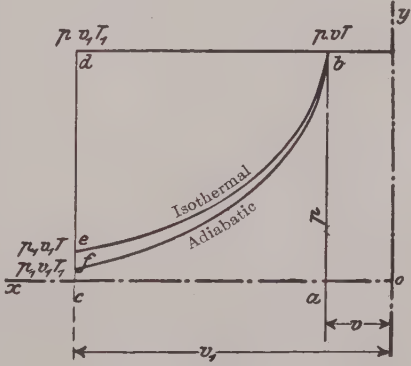

FIG. 699.

If during the expansion just so much heat is supplied as is required to maintain the temperature constant at T, the pressure p would have decreased, according to Boyle's law, eqs. (12) and (13), and at every stage of the expansion $p_1 = p\frac{v}{v_1}$. If at a number of points between a and c, ordinates are erected equivalent to the corresponding pressures, as determined from the above equation, and if the end points of these ordinates are connected by a smooth curve, the curve will be the *isothermal be.* The final condition at the point e is given by v_1, p_1, T.

Finally, if we assume that no heat whatever is supplied to the gas during the expansion from v to v_1, and that the enclosing walls are impermeable to heat so that the change proceeds without either supplying or withdrawing heat (adiabatic change), we may apply the laws of Poisson, as stated in eqs. (20) to (22). If the ordinates corresponding to any intermediate volumes are computed from these equations, it will be found that the pressures decrease much faster than they did for the isothermal line and that consequently the *adiabatic* line *bf* diverges more and more rapidly from the isothermal *be.* The end condition at the point f is defined by p_1, v_1, T_1.

If we keep in mind that the isothermal line *be* follows the law $\frac{p}{p_1} = \frac{v_1}{v}$, while the adiabatic line is determined by the equation $\frac{p}{p_1} = \left(\frac{v_1}{v}\right)^x$, it will be seen at once that the greater the exponent x, the more rapidly will the adiabatic approach the x-axis. If the value of the exponent is some other value than x, say $n \lessgtr 1$, the simplest way of constructing the curve is by the method of Brauer.[1] From the origin, Fig. 700, lay off the angle α below the x-axis, and compute the angle β for the y-axis according to the equation

$$1 + \tan\beta = (1 + \tan\alpha)^n. \qquad (31)$$

From z, the point fixing the original condition of the gas and hence the starting point of the curve, draw the vertical zs and the horizontal zw. Through the intersections s and w draw the oblique lines at 45° as shown, and from the points s_1 and w_1 so determined draw the next set of verticals and horizontals. Their intersection in z_1 determines the next point on the curve. Continuing this process determines the curve. The net-work of construction lines not only determines the polytropic line through the point z, but it is possible by their use to draw a series of such polytropics. A second curve is indicated in Fig. 700. The smaller the angle α is taken, the closer together will be the points of intersection on the curve, and the more accurate therefore will be the curve. Table 145 gives several practical values of α and β for exponents from $n = 1.1$ to 1.41.

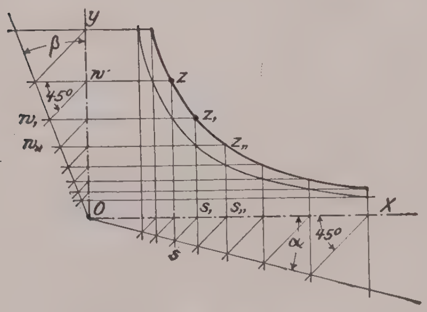

FIG. 700.

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[1] Z. d. V. D. I., 1885, p. 433.

TABLE 145

Exponent $n=$	1.10	1.15	1.20	1.25	1.30	1.35	1.41
Angle α ^{0}	$11.^{20}$	$11.^{20}$	$11.^{20}$	$14.^{05}$	$14.^{05}$	$14.^{05}$	$18.^{25}$
Angle β ^{0}	$12.^{35}$	$13.^{10}$	$13.^{50}$	$17.^{55}$	$18.^{40}$	$19.^{25}$	$26.^{30}$
Tan α	$0.2=1:5$	0.2	0.2	$0.25=1:4$	0.25	0.25	$0.33 =1:3$
Tan β	0.222	0.234	0.245	0.322	0.337	0.352	$0.497\sim 1:2$

For $n=1.0$ (isothermal line) the pressure-volume curve becomes a common hyperbola.

Instead of constructing the pressure-volume curve from given data we can of course go through the reverse process and from given pressure lines, as for instance those given by the engine indicator, determine the condition of the gas with respect to v, p, and T. Since the exponent in the equation for the adiabatic change is equal to $x=\frac{c_p}{c_v}$, we know at once that, if an actual pressure-volume *expansion line* for any given gas shows an exponent n different from x for this gas, the change was not adiabatic. If in such a case $n>x$, heat has been withdrawn during the change, while, if $n<x$, heat must have been supplied. On the other hand, the opposite relations hold for the *compression line.*

In practice, however, an expansion or compression line may appear to have an exponent $n=x$ and still not be a true adiabatic. This occurs when the amount of heat withdrawn from the gas during the change is just balanced by the amount supplied. Consequently if the expansion line of a gas engine indicator diagram shows an exponent equal to x, this by no means proves that the expansion has been adiabatic, but rather that a considerable amount of after-burning has taken place. The highly heated gases must during the expansion transfer a considerable quantity of heat to the cooling water, etc., which quantity can only be restored by after-burning if the expansion line appears adiabatic, that is, shows $n=x$. A curve of this kind has by some writers been called a *false adiabatic.*

From this standpoint we may summarize what is said above as follows: In all expansion lines for which $n<x$, that is, those that are situated *above* the theoretical adiabatic curve, the supply of heat is greater than the simultaneous withdrawal of heat, while the reverse is the case for all expansion lines for which $n>x$.

The deductions concerning the process of expansion made on the lines of Fig. 699, also apply without change to the compression of all perfect gases. The adiabatic compression line therefore rises more rapidly than the isothermal compression line starting from the same point, and for all lines lying above the isothermal the greater the exponent n in the equation $pv^n=\text{const.}$, the steeper the slope of the curve.

The exponent n for that part of a polytropic curve lying between the end points 1 and 2, Fig. 701, may be found from the equation

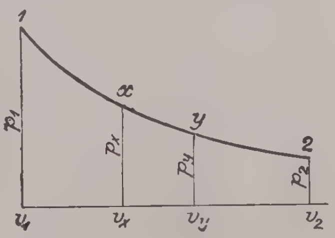

FIG. 701.

$$n=\frac{\log p_1-\log p_2}{\log v_2-\log v_1}, \qquad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \qquad (32)$$

in which p and v are as indicated in the figure. In this case n is considered constant for the entire curve. In fixing the points 1 and 2 care should be taken to see that they are located on the expansion line proper and not on the end of the combustion or the beginning of the exhaust line respectively. If it is desired to determine the variation of n along the curve, divide the latter into a sufficient number of parts (5 to 10) and from the values of p and v for the end points of each one of the partial lengths, say between the ordinates x and y, n may be computed from

$$n_{x/y}=\frac{\log p_x-\log p_y}{\log v_y-\log v_x}. \qquad (32a)$$

An accurate investigation of gas engine diagrams always calls for a subdivision of the expansion or compression lines, as above indicated, into as many parts as practicable, sometimes on an enlarged scale, because the exponent n, especially in the case of the expansion line, often changes very considerably. (See p. 38, Part I.)

6. Relations between Heat and Work, and the Mechanical Equivalent of Heat. A gas, expanding when heated, does external work, and conversely, doing work upon a gas, as compressing it, generates heat. There is a certain definite relation between the unit of work (foot-pound) and the unit of heat (British Thermal Unit) and it has been found experimentally that to generate mechanically one thermal unit requires the expenditure of mechanical energy equivalent to 778 ft.-lbs. Conversely, one B.T.U., when completely utilized, can do work equivalent to 778 ft.-lbs. Hence, 1 B.T.U. = 778 ft.-lbs. is the mechanical equivalent of heat, while 1 ft.-lb. $=\frac{1}{778}$ B.T.U. may be called the heat equivalent of work. The latter value is usually designated by A, so that

$$A=\frac{1}{778}=.001284 \text{ B.T.U.}$$

The first accurate determination of the mechanical equivalent was made by Joule, who found at that time $\frac{1}{A}\sim 772$ ft.-lbs. This figure is generally known even to-day as Joule's equivalent.[1]

Heat supplied to a body of gas has three effects: it increases the mobility of the molecules, which manifests itself on the one hand in greater amplitude of vibration (temperature) and on the other tends to change the relative position of or the distance between the atoms (internal energy), the two effects together transforming a certain amount of the heat into internal work. The third effect of supplying heat is to increase the volume of the gas or its pressure, and this represents external work. If the two parts of the internal work U are designated by W and J respectively, and the external work by L, we may write for an infinitesimal supply of heat that

$$dQ=A(dW+dJ+dL)=A(dU+dL). \qquad (33)$$

Eq. (33) expresses mathematically the *first law* of Thermodynamics. If for $dJ+dL$ we write dH, so that

$$dQ=A(dW+dH).$$

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[1] Regarding the uncertainty of this value, which varies from 772 to 782, see p. 2, Part I.

the first law may also be stated as follows: "The heat supplied to a body is partly used to raise the temperature and partly to increase the volume or the pressure of the same."

Assume now that we have one pound of any one of the perfect gases, with a volume at the start equal to v cu.ft., an absolute pressure of P pounds per sq.ft., and an absolute temperature of T degrees, enclosed by a piston in a non-conducting cylinder. If we next supply this body of gas with a certain quantity of heat equal to Q B.T.U., the effect will depend upon the action of the piston and may be as follows:

(*a*) *Volume Constant.* If the piston is held in its original position, (v=constant), the heat Q supplied will raise the temperature of the gas from T to T_1, and the pressure from P to P_1, according to the equation

$$T_1=T+\frac{Q}{c_v}=T\frac{P_1}{P} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad (34)$$

Conversely, having given T_1 and $P_1=P\frac{T_1}{T}$, we can find Q from [see also eqs. (27) to (29)],

$$Q=c_v(T_1-T)=\frac{c_v v}{R}(P_1-P) \text{ B.T.U.} \quad . \; . \; . \; . \; . \; . \; . \; . \quad (35)$$

Since v remains constant, the heat supplied did no external work.

(*b*) *Pressure Constant.* If the gas during a movement of the piston is furnished with a quantity of heat Q, sufficient to keep the pressure P constant, the temperature in the cylinder at the end of the expansion will be

$$T_1=T+\frac{Q}{c_p}=T\frac{v_1}{v}. \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad (36)$$

The external work done by the piston in its outward movement is equal to the pressure P into the change of volume, that is,

$$L=P(v_1-v) \text{ ft.-lbs., or } L=R(T_1-T) \text{ ft.-lbs.,} \quad . \; . \; . \; . \; . \; . \quad (37)$$

and the heat used will be

$$Q=c_p(T_1-T)=\frac{c_pP}{R}(v_1-v) \text{ B.T.U.} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \quad (38)$$

(*c*) *Temperature Constant.* If the supply of heat is so controlled that the temperature does not change during expansion, the final pressure will be

$$P_1=P\frac{v}{v_1} \text{ lbs. per sq.ft.}$$

The external work done will be

$$L=2.3026\ RT\log\frac{P}{P_1}=2.3026\ RT\log\frac{v_1}{v} \text{ ft.-lbs.,} \quad . \; . \; . \; . \; . \; . \; . \quad (39)$$

or

$$L=2.3026\ \ Pv\log\frac{v_1}{v}=2.3026\ P_1v_1\log\frac{v_1}{v} \text{ ft.-lbs.} \quad . \; . \; . \; . \; . \; . \; . \quad (39a)$$

Finally, the heat required is

$$Q=2.3026\ ART \log\frac{P}{P_1}=2.3026\ ART \log\frac{v_1}{v} \text{ B.T.U.} \quad . \quad . \quad . \quad . \quad . \quad (40)$$

The factor 2.3026 represents the ratio of natural to common logarithms, and disappears if the former are used in place of the latter.

(*d*) *No Heat Supplied during Expansion.* If the gas expands without gain or loss of heat (adiabatically) from the original state (p, v, T) to the volume v_1, the heat received is of course equal to zero. The change in pressure and volume is determined by eqs. (21) and (22). The work done is

$$L=\frac{c_v}{A}(T-T_1)=\frac{c_v}{AR}Pv\left[1-\left(\frac{v}{v_1}\right)^{x-1}\right] \text{ ft.-lbs.,} \quad . \quad . \quad . \quad . \quad . \quad . \quad (41)$$

or

$$L=\frac{Pv}{x-1}\left[1-\left(\frac{P_1}{P}\right)^{\frac{x-1}{x}}\right] \text{ ft.-lbs.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (41a)$$

The expressions developed above under (*c*) and (*d*), when used in the proper sense, also apply to the compression of a gas. As before, v, P, and T refer to the original and v_1, P_1, and T_1 to the final conditions. In this case, however, Q is the heat generated by the compression, that is the heat to be withdrawn, and it should be noted that it refers in all cases to one pound of gas.

7. Cycles. A cycle is defined as a series of successive pressure and volume changes in a body of gas, during which heat is supplied to and withdrawn from the latter in such order and quantity that the final condition of the gas is the same as the original. The pressure-volume lines of a cycle therefore form a closed polygon, but the individual enclosing lines may follow any law. Fig. 702 may serve as an example. Suppose that from the point 1 one pound of air (gas), having a volume v_1 cu.ft., absolute pressure P_1 lbs. per sq.ft. and an absolute temperature T, expands isothermally to the point 2. In order to maintain the temperature at T, we must supply a certain quantity of heat Q' B.T.U. From 1 to 2, the pressure has fallen from P_1 to P_2 lbs. per sq.ft. The external work done during this change was L' ft.-lbs., from eq. (39), represented in the diagram by the area $F'=1\ 2\ \text{II}\ \text{I}$. The heat supplied along 1–2 therefore is

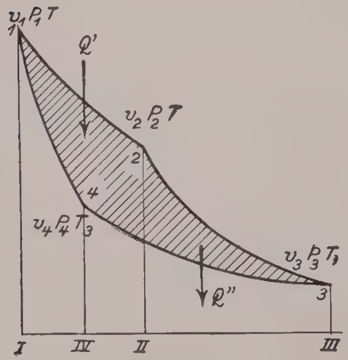

FIG. 702.

$$Q'=AL'=AF' \text{ B.T.U.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (42)$$

Next assume that the expansion from 2 to 3 is adiabatic, so that the external work must be done at the expense of internal energy in the gas. The temperature then falls from T to $T_{,}$, the pressure from P_2 to P_3 lbs. per sq.ft., while, according to eq. (41), the external work done is $L_{,}$ ft.-lbs., equivalent to the area $F_{,}=2\ 3\ \text{III}\ \text{II}$. The quantity of heat withdrawn from the gas to do the external work is

$$U_{,}=AL_{,}=AF_{,} \text{ B.T.U.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (43)$$

The quantity of heat Q supplied between 1 and 2 has during the expansion of the gas consequently done a total amount of work equal to $(L'+L_{,})$ ft.-lbs., represented by the area $\overline{1\;2\;3\;\mathrm{III}\;\mathrm{I}}$.

The piston next starts on its return stroke and first compresses the gas isothermally to the point 4. During this change a certain quantity of heat Q'' must be withdrawn from the gas in order to keep the temperature constant at T_3 in spite of the fact that P_3 increases to P_4. The work of compression L'' may be computed from eq. (39). It is represented by the area $F''=\overline{3\;\mathrm{III}\;\mathrm{IV}\;4}$ and its heat equivalent is

$$Q''=AL''=AF''\ \text{B.T.U.} \qquad (44)$$

If from the point 4 the compression proceeds adiabatically, without supply or withdrawal of heat, the gas will return to the original pressure P_1 and the original temperature T when the volume is again reduced to v_1, provided that the point 4 or the volume v_4 has been properly chosen. The external work $L_{,,}$ done upon the gas between 4 and 1 is determined from eq. (41). It is represented by the area $F_{,,}=\overline{4\;\mathrm{IV}\;\mathrm{I}\;1}$, and the internal heat (heat of compression) generated is expressed by

$$U_{,,}=AL_{,,}=AF_{,,}\ \text{B.T.U.} \qquad (45)$$

The hatched area $\overline{1\;2\;3\;4}=F$ represents the amount of external work done by the gas in passing through this cycle and is equal to

$$L=(L'+L_{,}-L''-L_{,,})\ \text{ft.-lbs.} \qquad (46)$$

Eq. (46) expresses the difference between the quantity of heat Q' supplied and the quantity Q'' withdrawn, hence

$$L=\frac{Q'-Q''}{A}=\frac{Q}{A}\ \text{ft.-lbs.} \qquad (47)$$

In eq. (46) since the external work done by the gas along $\overline{2\;3}$ is equal to the work done upon the gas along $\overline{4\;1}$,

$$L_{,}=L_{,,}.$$

Hence the total work done also equals

$$L=L'-L''. \qquad (47a)$$

From eq. (39), however,

$$L'=2.3026\ RT\log\frac{v_2}{v_1},\quad\text{and}\quad L''=2.3026\ RT_{,}\log\frac{v_3}{v_4};$$

therefore

$$L=2.3026\ RT\log\frac{v_2}{v_1}-2.3026\ RT_{,}\log\frac{v_3}{v_4}.$$

But for the adiabatic changes $\frac{v_2}{v_1}=\frac{v_3}{v_4}$, hence

$$L=2.3026\ R\ (T-T_{\prime})\log\frac{v_2}{v_1}=2.3026\ RT\frac{T-T_{\prime}}{T}\log\frac{v_2}{v_1}. \quad . \quad . \quad . \quad . \quad . \quad (48)$$

In this equation the factor $2.3026\ RT\log\frac{v_2}{v_1}$ corresponds according to eq. (40) to the quantity of heat supplied $\frac{Q'}{A}$, and substituting this we finally have

$$L=\frac{Q'}{A}\cdot\frac{T-T_{\prime}}{T}=\frac{Q'}{A}\left(1-\frac{T_{\prime}}{T}\right). \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (48a)$$

This equation expressed in words states: that the theoretical external work L done in a reversible cycle depends not only upon the quantity of heat Q' furnished to the cycle but also upon the ratio $\frac{T_{\prime}}{T}$ of the temperature limits.

The cycle just described is generally designated "reversible" because the individual changes could also be carried out in the reverse order with the same final result. In the special case above outlined, in which the limiting curves consist of two isothermals and two adiabatics, the cycle is the ideal (*Carnot cycle*) because for the given temperature limits it shows the theoretically best utilization of heat, that is the thermal efficiency η_t is the maximum. If, on the other hand, the heat has been supplied or withdrawn at varying temperature the cycle is called *polytropic* (according to Lorenz). Finally, we also distinguish "*closed*" and "*open*" cycles, depending upon whether the same body of working fluid always goes through the various changes of state or whether the working fluid is subject to constant renewals.

The *thermal efficiency* of a cycle is the ratio between the amount of heat transformed into external work divided by the total heat supplied to (or used by) the cycle. According to whether the heat supplied or the external work done refers to the theoretical cycle or to the actual indicator diagram, we distinguish

$$\left.\begin{array}{l}\text{Theoretical thermal}\\ \text{Indicated thermal}\end{array}\right\}\text{Efficiency}\left\{\begin{array}{l}\eta_t\\ \eta_i.\end{array}\right.$$

The ratio of these two corresponds to the efficiency of the combustion process and may also be considered to represent the "card factor," that is

$$\eta_g=\frac{\eta_i}{\eta_t}.$$

If as before Q' is the heat supplied, Q'' the heat discharged, so that $Q'-Q''=Q$ is the heat transformed into external work, we have in general

$$\eta_t=\frac{Q'-Q''}{Q'}=\frac{Q}{Q'}=\frac{AL}{Q'}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (49)$$

and for the ideal cycle in particular, as will easily be seen from eq. (48*a*),

$$\eta_t=\frac{T-T_{\prime}}{T}=1-\frac{T_{\prime}}{T}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (49a)$$

In the case of the Carnot cycle, therefore, the thermal efficiency depends only upon the range between the temperature limits, that is upon the fall of temperature.

If the indicated work of one cycle is L_i, while the heat supplied is Q, the indicated thermal efficiency will be

$$\eta_i=\frac{L_i}{L}=\frac{AL_i}{Q}. \qquad (50)$$

For the method of determining L_i, see p. 8, Part I.

8. "Heat Weight" and Entropy. From the derivation of eqs. (49) and (49*a*) it follows that for the ideal cycle we may write

$$\frac{Q'}{Q''}=\frac{T}{T_,}, \quad \text{hence also} \quad \frac{Q'}{T}=\frac{Q''}{T_,}. \qquad (51)$$

This ratio $\frac{Q}{T}$ of the quantity of heat supplied or discharged to its absolute temperature is usually called the "heat weight," following Zeuner's example. Eq. (51) may therefore be expressed in words as follows: "In an ideal cycle the heat weights supplied and discharged are equal to each other." This is the *second law of thermodynamics.*

Writing eq. (48*a*) in the form

$$Q=AL=\frac{Q'}{T}(T-T_,), \qquad (52)$$

and keeping in mind that $\frac{Q'}{T}$ is the heat weight and $T-T_,$ the temperature drop in the cycle, we may also state the second law as follows: "In an ideal cycle that part of the heat (Q) transformed into external work is equal to the product of heat weight by the temperature drop."

Since Q' represents the heat supplied and Q'' that discharged, so that the quantities are of opposite sign, we may write

$$\frac{Q'}{T}+\frac{Q''}{T_,}=0.$$

If now we assume that any given closed cycle is divided into an infinite number of elementary Carnot cycles, we must therefore have also for the summation of all these elementary cycles that

$$\Sigma\frac{Q}{T}=0 \quad \text{or} \quad \int\frac{dQ}{T}=0.$$

Clausius has given the expression $\int\frac{dQ}{T}$ the name "Entropy." With this conception the second law of thermodynamics states that "In any given closed cycle the entropy (sum of the heat weights supplied and discharged) is equal to zero."

By plotting entropy with temperature we obtain the so-called *entropy* diagram which is an invaluable aid in the study of the operation of internal-combustion engines (see p. 10 and 40, Part I).

II. Fundamental Principles of Thermochemistry

1. Atoms and Molecules and their Weights. A given body may be divided both chemically and mechanically. The infinitesimally small particles of an elementary material that can be divided no further either mechanically or chemically are known as *atoms.* It is assumed that in any given material these atoms are closely associated in certain groups, and such smallest possible atom-groups into which the material can be subdivided without affecting its chemical constitution, are called *molecules.*[1] Expressing this explanation in other words we may say that a molecule is the smallest weight necessary for the existence of a substance, while the atom is the smallest conceivable weight with which the chemical elements form combinations.

The atoms of the various elementary materials have a weight, called *atomic weight* (e), which, although impossible of direct determination, is definite and fixed. This weight is usually referred to that of hydrogen (the lightest element) as unity, and therefore expresses how many times the atom of a given body is heavier than the atom of hydrogen. The atomic weight of oxygen is $e=15.95$. If this figure is taken at 16.0, the atomic weight of hydrogen becomes $e=\frac{16}{15.95}=1.0032$. For practical reasons chemists have of late years found it better to make computations with oxygen $e=16.0$ than with hydrogen $e=1.0$.

The sum of the weights of all of the atoms of a molecule is known as the *molecular weight* (m). Since one molecule (H_2) of hydrogen consists of two atoms, its molecular weight is $m=2$. If based upon oxygen, however, $(O_2)=32$, the molecular weight of hydrogen is $m=\frac{32}{15.95}=2.0064$. Chemical computations are usually made with such units as the gram-molecule or the gram-molecule-volume, the numerical values for which are given below.

The number of molecules contained in a given volume of a body constitutes its *mass,*[2] while the relation between mass and volume is the density, (δ) of the material. According to the law of Avogadro, equal volumes of all gases at the same pressure (density) and temperature contain the same number of molecules. From this we derive the following important conclusions:

(*a*) The molecule-volume of all gaseous bodies is the same.

(*b*) The density of all gaseous bodies is in direct proportion to their molecular weight m.

In general the density of gases is referred to that of air as unity, that is δ expresses how many times heavier a given gas is than an equal volume of air under the same conditions.

In conformity with the law of Avogadro, above stated, it has been found that the pound-molecule-volume is equal to 358.17 cu.ft. Mathematically expressed this means that, if m represents the molecular weight as before, the volume of one pound of any of the perfect gases under standard conditions, the so-called *specific volume* v, is equal to

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[1] To get a clear conception of these statements, a gaseous body may be considered as a dust-cloud, in which the individual dust grains are made up of the finest elementary particles of some elementary substance. The dust grains then represent the molecules and their constituent particles the atoms.

[2] Mass (M) and weight (G) of a body are proportional to each other, but not equal, since $M=\frac{G}{32.2}$. The mass of a body is constant, but the weight depends upon the acceleration due to gravity, that is upon the geographic location.

$$v=\frac{358.17}{m} \text{ cu.ft.}, \qquad (1)$$

and conversely that the weight of one cubic foot of any of the perfect gases under standard conditions is equal to

$$\gamma=\frac{1}{v}=\frac{m}{358.17} \text{ lbs.} \qquad (2)$$

From this equation the last column of the following table has been computed.

TABLE 146

CHEMICAL CONSTANTS FOR PERFECT GASES

Kind of Gas.	Atom Number.	Atomic Weight.	Molecular Weight.			Density Referred to Air=1.0	Weight per Cubic Foot at 32° and 14.7 lbs.
			Approximate, $H_2=2.$	Accurate for			
				$H_2=2.$	$O_2=32.$		
	n	e	m	m	m	δ	γ
Hydrogen .... H_2	2	1	2	2.0	2.006	.0693	.005588
Oxygen .... O_2	2	16	32	31.9	32.0	1.1056	.0892
Nitrogen .... N_2	2	14	28	28.00	28.00	.971	.0783
Carbon monoxide .... CO	2	....	28	27.93	28.00	.967	.0781
Nitric oxide .... NO	2	....	30	....	30.40	1.031	.0838
Carbon dioxide .... CO_2	3	....	44	43.87	44.00	1.519	.12262
Air, dry	....	....	29	28.88	....	1.0	.08069
Water vapor .... H_2O	3	....	18	....	18.0	.623	.0503
Acetylene .... C_2H_2	4	....	26	25.94	26.01	.915	.0724
Methane .... CH_4	5	....	16	15.97	16.02	.554	.0446
Ethylene .... C_2H_4	6	....	28	27.94	28.02	.974	.0781
Alcohol .... C_2H_6O	9	....	46	....	46.03	1.601	.13072
Propylene .... C_3H_6	9	....	42	41.91	....	1.451	.1171
Benzol .... C_6H_6	12	....	78	77.82	78.04	2.695	.2177

For carbon (C_2), $e=11.965\sim12$; $m=23.93\sim24$; $\delta=.820$; and $\gamma=.0668$.

2. Elements and their Combinations and Symbols. Substances that can not be subdivided into simpler ones are called *elements*. Two or more elements may unite to form one or more chemical compounds either by the direct combination of their molecules or by a rearrangement resulting in what would correspond to the substitution of the atoms of one of the elements in the molecule of the other.

Chemical combinations are entered into with ease, with difficulty, or not at all, depending upon the chemical relationship (affinity) of the elements concerned. If the chemical combination takes place it always occurs without loss of matter (weight), but is usually accompanied by a decrease in the number of molecules and hence contraction of specific volume.

Elements and chemical combinations are designated by the first letter of their (usually Latin) names, the number of atoms per molecule being at the same time defined by numerical subscripts. The chemical symbols of all of the gases likely to be used in the gas engine industry are given in Table 146. The symbol for hydrogen, for instance, is H_2, which states that the molecule of hydrogen consists of two atoms. Leaving out the atom number (in this case 2) gives to the symbol a double meaning, the letter H then not only signifies hydrogen in general but also a definite chemical quantity, that is, one atom of hydrogen, which in chemical relations has only one definite value. The same applies of course to the chemical symbols of all of the elements.

One atom of carbon (C) combines with one atom of oxygen (O) to form a diatomic molecule of carbon monoxide (CO). In this case the combination formula (CO) lacks the atom number because the number of atoms in the union is definitely stated by the linking of the two symbols. The symbol for carbon dioxide is CO_2, which states that it is a combination of one atom of carbon with two of oxygen, and one molecule of CO_2 therefore consists of three atoms. The same number of atoms is contained in water vapor (H_2O), ammonia (NH_3) has four atoms, methane (CH_4) five, benzol (C_6H_6) twelve, etc. If a group of similar combinations is to be indicated by one general symbol, the atom number is replaced by the subscript n. For instance, C_nH_{2n} designates a combination of n atoms of C with $2n$ atoms of H. In the case of ethylene (C_2H_4), therefore, $n=2$, while for hexylene (C_6H_{12}), $n=6$, both belonging to the group of hydrocarbons designated collectively by C_nH_{2n}. The symbol C_nH_{2n+2} states that a molecule contains n atoms of C to $2n+2$ atoms of hydrogen, as for instance methane, CH_4.

The determinations of volumes or weights of chemical combinations is done by means of their molecular weights, the same as for the elements. If the molecular weight m is not given it may be easily computed from the atomic weights e of the elements in the combination.

Examples. Carbon monoxide has a molecular weight equal to 27.93, consequently 1 cu.ft. from eq. (2) will weigh

$$\gamma=\frac{27.93}{358.17}=.0781 \text{ pounds.}$$

The same result would have been obtained by starting with the weight of a standard cubic foot of hydrogen, and remembering that the densities of the perfect gases are to each other as their molecular weights, as follows:

$$\gamma=.005588\times\frac{27.93}{2}=.0781 \text{ pounds.}$$

One molecule of carbon dioxide is composed of one atom of C (whose atomic weight is $\frac{23.93}{2}=11.965$) and two atoms of O (whose atomic weight is $\frac{31.90}{2}=15.95$); its molecular weight therefore is (sum of the weights of the three atoms) equal to $11.965+2\times15.95=43.865$. One cubic foot of CO_2 consequently weighs

$$\gamma=\frac{43.865}{358.17}=.12262 \text{ pounds,}$$

and its density is (air=1.0),

$$\delta=\frac{.12262}{.08069}=1.519.$$

One molecule of ethylene consists of

2 atoms of C, of weight $=2\times 11.965=23.93$,

and 4 " " H, " " $=4\times 1.0\quad=\ 4.00$,

hence the molecular weight of $C_2H_4=27.93$

and 1 cu.ft. of C_2H_4 weighs $\frac{27.93}{358.17}=.0781$ lbs. at 32° F. and 14.7 lbs. abs. pressure.

Any hydrocarbon combination of general formula C_nH_n has a molecular weight of

$$M_0=11.96n+1n=(11.96+1)n\sim 13n \quad . \ . \ . \ . \ . \ . \ . \ . \quad (3)$$

in which 11.96 = atomic weight of C and 1.0 = that of H. For the so-called heavy hydrocarbons of general formula C_nH_{2n} we have similarly

$$M_0=11.96n+2n=13.96n\sim 14n, \quad . \ . \ . \ . \ . \ . \ . \ . \ . \quad (4)$$

and the density of the latter group referred to air as unity will be

$$\delta=\frac{14n}{.08069}. \quad . \ . \ . \ . \ . \ . \ . \ . \ . \ . \ . \quad (5)$$

If any chemical combination is composed of several elements E_1, E_2, $E_3, \ldots$, whose atomic weight is e_1, e_2, $e_3, \ldots$, and the number of atoms in which is n_1, n_2, $n_3, \ldots$, respectively, the molecular weight of the combination is

$$m=n_1e_1+n_2e_2+n_3e_3+\ldots=\Sigma(ne). \quad . \ . \ . \ . \ . \ . \ . \ . \quad (6)$$

3. Combining Weights and Volumes, Atomic Heat, Diffusion and Dissociation. According to the law of conservation of matter, the elements and their weights remain unchanged during any chemical change. A combination of different gases therefore has exactly the same weight as the sum of the weights of the constituent gases. Chemical reactions in general only take place in definite weight proportion, and in the case of the gases usually also in simple volume proportions. The smallest weight (or volume) with which any given element will still combine with another element is known as its *combining weight* (*or volume*).

In gas engine practice it is often most important to know the volume ratios in which given gases combine. This is most easily determined by writing down the chemical combination formulas of the gases concerned, the number of molecules of each gas in the combination will then give the volume, as the following examples show:

(*a*) Combustion of C to CO:

1 molecule of C_2 + 1 molecule of O_2 = 2 molecules of CO,

hence

1 volume of C + 1 volume of O = 2 volumes of CO

(*b*) Combination of hydrogen and chlorine gas:

1 molecule of H_2+1 molecule of Cl=2 molecules of HCl,

hence

1 volume of H+1 volume of Cl=2 volumes of HCl (hydrochloric acid).

(*c*) Combustion of H to water vapor:

2 molecules of H_2+1 molecule of O_2=2 molecules of H_2O,

therefore

2 volumes of H+1 volume of O=2 volumes of H_2O.

(*d*) Combination of C and H to form methane, CH_4:

1 molecule of C_2+4 molecules of H_2=2 molecules of CH_4.

therefore

1 volume of C+4 volumes of H=2 volumes of CH_4.

The term *atomic heat* means the product of the specific heat of an element by its atomic weight. The *atomic heat* is a nearly constant quantity, having about the same numerical value (6.0 to 6.5) for all of the elements. It follows from this that equal numbers of atoms require the same quantity of heat to raise the temperature the same amount. *Diffusion* is not a chemical combination but is simply the process by which the molecules of liquids or gaseous bodies form perfectly uniform mixtures, there being no chemical reaction of the elements (or molecules) upon each other. In atmospheric air for instance we find diffused nitrogen and oxygen. Air is therefore not a chemical compound. Diffusion of gases is unlimited, as long as the gases do not react chemically, but it takes place in different gases with varying rapidity. The measure of the power of diffusion is called the *diffusion coefficient*, which represents the "diffusion length," that is, the distance traveled by one gas in another in one second. The experimental data available on this point is rather scarce and somewhat unreliable as far as practice, as distinguished from laboratory work, is concerned. Thus for carbon dioxide and air at 32° and 29.92″ Hg., various tests show the diffusion coefficient to vary from .053 to .059 in. per second. In general, the lower the density of the gases and the greater the mobility of the molecules, the more rapid the diffusion. For that reason an increase of temperature and mechanical agitation will aid the process, while compressing the gas retards it on account of the increase in density. According to the investigations of Loschmidt, the diffusion coefficient varies inversely as the density (pressure) of the gases and nearly directly as the square of the absolute temperatures.[1] In compressing a gas, however, although the increase of presssure decreases the diffusion coefficient, the simultaneous decrease of volume decreases the diffusion distance in the same ratio and hence the latter effect balances the former. The result therefore is that *pressure or density has no influence upon the diffusing power of gases.*

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[1] Meyer, Kinetische Theorie der Gase, p. 250.

It has been shown that the relative proportion of the gases in the mixture has no sensible influence upon the diffusion process.

The field of every chemical action is limited by an upper and a lower temperature. Either above the upper limit or below the lower limit, chemical reaction ceases, and the combinations formed decompose into their constituents, at least above the upper limit. This decomposition process, which thus depends upon the temperature, is known as *dissociation,* and the temperature at which it makes its appearance is called the *dissociation limit.* At temperatures exceeding 3600° F., for instance, water vapor (H_2O) breaks up into hydrogen and oxygen, and carbon dioxide into carbon monoxide and oxygen. Since the development of heat in an internal-combustion engine is the result of chemical reaction, it follows that the combustion temperatures of the mixtures must not be allowed to exceed the upper dissociation limit, otherwise the development of heat would be interrupted at the upper limit until the temperature has again fallen below it. An increase of pressure usually raises the upper dissociation limit, a drop of pressure consequently lowers it.

Every combination of oxygen with other elements is known by the general term *oxidation,* and is generally accompanied by a decomposition, development of heat, or other phenomena. The most important oxidation process is combustion. The reverse process, that is, the extraction of oxygen from oxygen compounds, is known as *reduction.* While oxidation produces heat, reduction absorbs it. Among the practical reducing agents are carbon, hydrogen, certain hydrocarbon compounds, etc. Carbon (C) for instance, is oxidized to CO_2 (carbon dioxide). If now the CO_2 formed is led through an incandescent layer of carbon, it is reduced according to the equation $C+CO_2=2CO$ (carbon monoxide). These alternate actions are of the greatest importance in the production of producer gas (see p. 593).

4. Combustion. Most chemical reactions are accompanied by a development of heat, although this may not be easily noticeable in some cases. During the combination of oxygen with certain related elements, however, notably hydrogen and carbon, the accompanying chemical energy is so completely transformed into heat that the materials concerned either glow or ignite, thus making the process visible to the eye and perceptible to the touch. This phenomenon is known as *combustion.* Oxygen combinations of this kind only form in definite atomic or weight ratios, which may be expressed by chemical equations as follows:

$H_2+O=H_2O$, that is, two atoms of hydrogen combine with one atom of oxygen to form one molecule of water vapor containing three atoms.

$C+O_2=CO_2$, that is, one atom of carbon combines with two of oxygen to form one molecule of carbon dioxide containing three atoms.

$C+O=CO$, that is, one atom of carbon combines with one of oxygen to form one molecule of carbon monoxide containing two atoms.

After substituting the atomic weight for the atomic number (see table 146, p. 581) in above relation, these may be written:

$$\left.\begin{array}{l}2 \text{ lbs. of } H+16 \text{ lbs. of } O=18 \text{ lbs. of } H_2O\\ 1 \text{ lb. of } H+8 \text{ lbs. of } O=9 \text{ lbs. of } H_2O\end{array}\right\} \text{Molecular weight of } H_2O \text{ is therefore } m=18$$

$$\left.\begin{array}{l}12 \text{ lbs. of } C+32 \text{ lbs. of } O=44 \text{ lbs. of } CO_2\\ 1 \text{ lb. of } C+\tfrac{32}{12} \text{ lbs. of } O=3.666 \text{ lbs. of } CO_2\end{array}\right\} \text{Molecular weight of } CO_2 \text{ is therefore } m=44$$

$$\left.\begin{array}{l}12 \text{ lbs. of } C+16 \text{ lbs. of } O=28 \text{ lbs. of } CO\\ 1 \text{ lb. of } C+\tfrac{16}{12} \text{ lbs. of } O=2.333 \text{ lbs. of } CO\end{array}\right\} \text{Molecular weight of } CO \text{ is therefore } m=28$$

The combustible elements are not usually available in the pure form, but are found in combinations or mixed with other and indifferent elements.

In order to start combustion there is required a certain minimum temperature known as the *ignition temperature.* The latter may be produced chemically (by means of glowing or flaming materials, etc.) or mechanically (by means of pressure, friction, etc.). The temperature of ignition for any given element apparently varies with attendant circumstances, and is therefore not definitely fixed. For hydrogen it is about 1020° F. and for methane and carbon monoxide in the neighborhood of 1200° F. The ignition temperature for carbon seems to be about midway between these values. The low ignition temperature of hydrogen explains why gas engine fuel mixtures high in hydrogen can stand only a moderate compression. The *combustion temperature* is of course much higher than the ignition temperature, although the former, on account of the interruption of the process of oxidation caused by dissociation, practically never reaches the theoretically possible value. Hydrogen gas, for instance, if burned in pure oxygen without dissociation, would show a combustion temperature of from 11 700 to 12 600° F., while carbon under the same conditions would even reach 18 000° F. In an actual case, on the other hand, neither combustion, on account of dissociation, shows much over 3600° F. In general, a non-luminous flame burns at considerably higher temperature than one that is luminous. The reason for this is found in the heat consumed by the solid, carbonaceous particles which give the light in the flame, and their incomplete combustion. A *non-luminous flame* is produced when the fuel gas before its ignition is furnished with so much air (oxygen) that its combustion is instantaneous and complete (Bunsen burner).

The quantity of heat which one pound of the combustible produces during the union with oxygen is called the *heating value* and differs considerably for the various fuels. Thus one pound of hydrogen, when completely burned, will develop 62 000 B.T.U., one pound of carbon 14 544 B.T.U., and one pound of sulphur 3966 B.T.U.

A fuel is burned completely when in the oxidation process all carbon is changed to carbon dioxide, all hydrogen to water vapor, and all sulphur to sulphur dioxide. If the combustion does not attain this "maximum degree of oxidation" the heat developed may be considerably less. Thus carbon, when burned to carbon monoxide instead of carbon dioxide, develops only 4446 B.T.U. per lb. Chemically this interrelation between heat of combustion and degree of oxidation is indicated as follows:

$H_2+O=H_2O$ (liquid)	+68.90	cal. per gr.-mol.	or	61 520	B.T.U. per lb.
$C+O_2=CO_2$	+96.96	"	"	14 544	"
$C+O=CO$	+29.56	"	"	4446	"

Any combustion is proved complete when the products of combustion, both solid and gaseous, show no combustible or unburned constituents. Since in gas engines only gaseous fuels (hydrogen, carbon monoxide, and hydrocarbon compounds) are burned, not in oxygen, but in air, the exhaust gas must consist of carbon dioxide, water vapor, nitrogen, and excess oxygen.

Theoretically one atom of C requires two atoms of O for its complete combustion, while one atom of H only requires one-half atom of O. The theoretically least amount of oxygen required to completely burn one pound of each combustible may then be found from the atomic weights as follows:

For Carbon, C

$$G=\frac{2\times 16}{12}=\frac{8}{3}=2.667\text{ lbs.,}$$

For Hydrogen, H

$$G_s=\frac{.5\times 16}{1.0}=8\text{ lbs.}$$

Since one pound of air contains .235 pound of oxygen, the least amount of air required for the complete combustion of C_2 and H_2 will be

$$G_l=\frac{2.667}{.235}=11.35\text{ lbs. for C.}$$

and

$$G_l=\frac{8}{.235}=34.04\text{ lbs. for H.}$$

If the fuel does not consist of a single element as above, but is composed of combinations of the fuel gases mentioned, or combinations of these with other gases, each of the latter must be considered in computing the necessary oxygen or air required according to its atomic weight *e*, or its atomic number *n*. Thus one pound of a gas having the composition $(C_{n_1}H_{n_2}O_{n_3}N_{n_4})$ would require a weight of oxygen equal to

$$s=\frac{2n_1e_3+.5n_2e_3-n_3e_3}{m}=\frac{32n_1+8n_2-16n_3}{m}\text{ lbs.,} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (7)$$

in which *m* is the molecular weight of the gas. The burned gases resulting from the complete combustion of this gas will consist of

$$k=\frac{n_1(e_1+2e_3)}{m}=\frac{44n_1}{m}\text{ lbs. of }CO_2, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (8)$$

and

$$w=\frac{n_2(e_2+.5e_3)}{m}=\frac{9n_2}{m}\text{ lbs. of }H_2O, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (9)$$

together with the nitrogen and the excess of oxygen in the air used for combustion, both of which remain uncombined. By means of these equations the data of Table 147 for the main constituents of illuminating gas has been computed.[1] The average illuminating gas quoted in the last row of the table has the following composition by weight: $.54CH_4+.10C_2H_4+.08C_4H_8+.05H_2+.15CO+.08N_2$.

By use of the columns for *s*, *k* and *w*, in this table, the theoretical amount of oxygen required by an illuminating gas, each pound of which contains say G_1 lbs. CH_4, G_2 lbs. C_2H_4, G_3 lbs. C_4H_8, . . ., may be directly found by substituting in the formula

$$\text{Oxygen}=\Sigma(Gs)\text{ pounds.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (10)$$

Similarly, carbon dioxide and water vapor contained in the burned gas may be determined from

$$\text{Carbon dioxide}=\Sigma(Gk)\text{ pounds} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (11)$$

and

$$\text{Water vapor}=\Sigma(Gw)\text{ pounds.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (12)$$

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[1] Mainly following Zeuner, Thermodynamik, 2d ed., I., p. 398.

TABLE 147

COMBUSTION DATA FOR THE PRINCIPAL GASES

Kind of Gas.	Lower Heating Value, H_u		Theoretically Required.			Products of Combustion per lb. of gas Contain		Specific Heat of the Gas		Gas Constant
			Oxygen (s)	Air (l)						
	per lb., B.T.U.	per cu.ft., B.T.U.	per lb., lbs.	per lb. lbs.	per cu.ft., cu.ft.	Carbon Dioxide (k) lbs.	Water Vapor (w) lbs.	c_p	c_v	R
Hydrogen, H_2	51696	289	8.0	34.0	2.38	....	9.0	3.409	2.412	770.2
Nitrogen, N_2	....	....	....	....	....	....	....	.244	.173	55.08
Carbon monoxide, CO	4334	338	.57	2.42	2.38	1.57	....	.245	.174	55.21
Methane, CH_4	21385	952	4.00	17.00	9.52	2.75	2.25	.593	.468	96.47
Ethylene, C_2H_4	20025	1584	3.43	14.55	14.29	3.14	1.29	.404	.333	55.19
Butylene, C_4H_8	19508	3055	3.43	14.55	28.25	3.14	1.29	.404	.333	27.51
Average illuminating gas *	18200	700	3.26	13.85	6.66	2.29	1.90	.619	.473	110.87

* This average quality of illuminating gas is much better than that found in the United States, where the average B.T.U. per cubic foot is about 575.

According to the so-called combination formulæ, one pound of fuel, consisting of C pounds of carbon, H pounds of hydrogen, O pounds of oxygen, and S pounds of sulphur, theoretically requires for its complete combustion

$$\frac{\frac{8}{3}\,C+8\,H+S-O}{.23}\ \text{lb.,}\quad\text{or}\quad\frac{\frac{8}{3}\,C+8\,H+S-O}{.0187}\ \text{cu.ft. of air.} \quad . \quad . \quad . \quad . \qquad (13)$$

If the resulting products of combustion consist of $k\%$ carbon dioxide, $s\%$ oxygen, and $n\%$ nitrogen, by volume, the ratio of the air actually used to that theoretically required will be

$$\text{Air excess coefficient}=\frac{21}{21-79\frac{s}{n}}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (14)$$

If instead of the weight per cent, the volume per cent composition of a hydrocarbon compound $C_{n_1}H_{n_2}$ is given, one cubic foot of the gas will require

$$\mathfrak{S}=\left(n_1+\frac{n_2}{4}\right)\ \text{cu.ft. of oxygen.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (15)$$

Example. The derivation and use of (15) is best explained by a problem. Suppose that the given hydrocarbon gas is C_2H_4. This makes $n_1=2$ and $n_2=4$. The gas may be considered made up as follows:

$$C_2+2H_2=C_2H_4,$$

$$1\ \text{volume of}\ C+2\ \text{volumes of}\ H=1\ \text{volume of}\ C_2H_4,$$

hence the combustion of 1 cu.ft. of C_2H_4 may be considered equivalent to the combustion of 1 cu.ft. of C (gas) and of 2 cu.ft. of H.

(*a*) Combustion of 1 cu.ft. of C:

$$C_2+2O_2=2CO_2,$$

$$\text{1 volume of C}+\text{2 volumes of O}=\text{2 volumes of }CO_2,$$

hence 1 cu.ft. of C requires 2 cu.ft. of O.

(2) Combustion of 2 cu.ft. of H:

$$2H_2+O_2=2H_2O,$$

$$\text{2 volumes of H}+\text{1 volume of O}=\text{2 volumes of }H_2O,$$

hence 2 cu.ft. of H require 1 cu.ft of O.

The combustion of 1 cu.ft. of C_2H_4 then requires

$$\mathfrak{S}=2+1=3=\left(n_1+\frac{n_2}{4}\right)\text{ cu.ft. of oxygen,}$$

which proves eq. (15).

From eq. (15) air required for the combustion will then be

$$\mathfrak{L}=\frac{\mathfrak{S}}{.213}=4.695\left(n_1+\frac{n_2}{4}\right)\text{ cu.ft.} \tag{16}$$

The combustion formula for this gas will read

$$C_{n_1}H_{n_2}+\left(n_1+\frac{n_2}{4}\right)O=n_1CO_2+\frac{n_2}{2}H_2O,$$

or put into words: One volume of $C_{n_1}H_{n_2}$ will burn with $\left(n_1+\frac{n_2}{4}\right)$ volumes of oxygen to form n_1 volumes of CO_2 and $\frac{n_2}{2}$ volumes of H_2O, water vapor.

The method of determining the numerical factors n_1 and $\frac{n_2}{2}$ for carbon dioxide and water vapor in the above statement will be clear from the example above given.

Finally, as a general case, suppose another gas has the following composition by volume, the letter x with the proper subscript representing the per cent by volume of the particular gas. Let the composition be

$$x_1CO+x_2H_2+x_3CH_4+x_4C_2H_4+x_5C_2H_2+x_6O_2+x_7N_2+x_8CO_2+x_9H_2O=1\text{ cu.ft.}$$

Then the amount of oxygen theoretically required by one cubic foot of this gas will be,

$$\mathfrak{S}=\left[\frac{x_1+x_2}{2}+2x_3+3x_4+2.5x_5-x_6\right]\text{ cu.ft.} \tag{17}$$

As before, the air theoretically required per cubic foot of gas will be

$$\mathfrak{L}=4.695\mathfrak{S}=4.695\left[\frac{x_1+x_2}{2}+2x_3+3x_4+2.5x_5-x_6\right]\text{ cu.ft.} \tag{17a}$$

If now $\mathfrak{L}'$ cubic feet of air are actually supplied per cubic foot of gas burned ($\mathfrak{L}' > \mathfrak{L}$), the products of combustion resulting will be

$$CO_2 = [x_1 + x_3 + 2x_4 + 2x_5 + x_8] \text{ cu.ft.}; \quad \ldots \quad (18)$$

$$H_2O = [x_2 + 2x_3 + 2x_4 + x_5 + x_9] \text{ cu.ft.}; \quad \ldots \quad (19)$$

$$N = [.79\mathfrak{L}' + x_7] \text{ cu.ft}; \quad O = [.21\mathfrak{L}' - \mathfrak{S}] \text{ cu.ft.} \quad \ldots \quad (20)$$

These equations represent the cubic feet of burned gases actually produced per cubic foot of fuel gas. The results may be easily reduced to per cent composition by volume if desired.

The total volume of burned gases, reduced to the original pressure and temperature is usually smaller, never greater, than the original volume of the gases before combustion. The same holds true concerning the number of molecules before and after combustion. Contraction therefore occurs. For a gas of the general formula C_nH_n the amount of contraction, according to Mollier, may be expressed by

$$\Delta V = \left(1 - \frac{n}{4}\right) \text{ parts by volume,} \quad \ldots \quad (21)$$

while for the second and more complex composition above given it is approximately equal to

$$\Delta V = \frac{CO + H_2 + C_2H_2}{2} \text{ parts by volume.} \quad \ldots \quad (21a)$$

It is here assumed that the water in the products of combustion really escapes in the shape of vapor, which is of course always the case in gas engines.

In this connection it should be noted that the *heating value* of a fuel containing hydrogen depends upon the final condition of the water vapor (water of combustion) concerned in the combustion. If the burned gases leave at a temperature exceeding 212° the water vapor carries with it, and renders unavailable as far as any attempt at utilization is concerned, the latent heat of vaporization. The heat of combustion developed is thus reduced to the so-called *lower* heating value (H_u). If on the other hand the products of combustion before escaping are brought to a temperature below 212°, the water vapor is condensed during the time that the heat is being abstracted (used) and the latent heat of vaporization thus returned increases the available heat of combustion to the *higher* heating value (H_0). Since one pound of water vapor under average conditions, say when cooled to 65° F., gives up about 1100 B.T.U., the difference between these two heating values amounts to

$$H_0 - H_u = 1100w, \quad \ldots \quad (22)$$

in which w is the weight of water resulting from the combustion of 1 lb. or 1 cu.ft. of fuel.

As far as gas engine practice is concerned, the lower heating value is always the one used, because the latent heat of the water vapor does not become available for transformation into work except at lower temperatures, and its value for this purpose is consequently small.[1]

To compute the heating value of solid and liquid fuels the following combination formula (DuLong's formula), with certain restrictions applicable also to gases, may be used.

$$H_u = \left[14500\,\mathrm{C} + 52200\left(\mathrm{H} - \frac{\mathrm{O}}{8}\right) + 4500\mathrm{S} - 1100\mathrm{H_2O}\right] \text{ B.T.U. per pound.} \quad . \quad (23)$$

In this equation, C, H, O, and S represent the part by weight of the elements mentioned, while H_2O is the water contained originally in 1 lb. of the fuel.

If 1 lb. or 1 cu.ft. of a gas mixture consists of t_1, t_2, t_3 parts by weight or by volume of the individual gases and the heating values of these gases are H_1, H_2, H_3 . . ., respectively, the heating value of 1 lb. or 1 cu.ft. of the mixture will be expressed by

$$\mathrm{H}_m = \Sigma(t\mathrm{H}) \text{ B.T.U.} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad (24)$$

If a fuel gas having a heating value of H B.T.U. is mixed with l parts by weight or by volume of air, the heating value of the resulting mixture will be

$$h = \frac{\mathrm{H}}{1+l} \text{ B.T.U. per lb. or per cu.ft.} \quad . \; . \; . \; . \; . \; . \; . \; . \quad (25)$$

The fuels in common use in gas-engine practice consist very largely of the hydrocarbon groups C_nH_{2n+2} or C_nH_{2n}. The heating value of such combinations is very simply determined from the heating values of their constituents C and H. The compound C_nH_n consists of n atoms of C and n atoms of H. Since the atomic weight of C is 12 and of H is 1, we may say that one pound-molecule of C_nH_n weighs $12n+n=13n$ lbs. Now 1 lb. of C will develop 14500 B.T.U., and 1 lb. of H, 52500 B.T.U. Hence the heating value of the combination will be

$$H_u = \frac{(12\times 14500)n + (1\times 52200)n}{(12+1)n} = \frac{(174000+52200)n}{13n} = 17400 \text{ B.T.U. per lb.} \quad . \; . \; . \quad (26)$$

It will be noted that the atom number finally cancels out, so that every hydrocarbon compound of the form C_nH_n will develop 17400 B.T.U. per lb. Since in this case the molecular weight of the combination is $13n$, the weight of a cu.ft. of C_nH_n will be

$$\gamma = \frac{13n}{358.17} \text{ lbs.,}$$

and the heating value of C_nH_n is consequently also

$$\mathrm{H}_u = 17400 \times \frac{13n}{358.17} = 632n \text{ B.T.U. per cu.ft.}$$

Example. For benzol (C_6H_6), $n=6$, hence the heating value is 17400 B.T.U. per lb., or $632\times 6=3792$ B.TU. per cu.ft.

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[1] This particular point was for a long time a much discussed question. See the thorough treatment given by E. Meyer in the Z. d. V. D. I., 1899, p. 326, etc., also same, 1903, p. 632.

In the same way the heating values of the heavy hydrocarbons C_nH_{2n} may be computed. In this case there are two atoms of H for every atom of C in the combination. Hence the pound-molecule weighs $12n+2n=14n$ lbs. Proceeding as above for C_nH_n, we finally get for C_nH_{2n}

$$H_u=19\,880 \text{ B.T.U. per lb.}, \qquad (27)$$

and

$$H_u=19880\times\frac{14n}{358.17}=778n \text{ B.T.U. per cu.ft.} \qquad (27a)$$

(*Note.* It should be distinctly stated that the heating values for the hydrocarbons computed as above do not in many cases agree with calorimetric determinations, as may be seen from the table following. The reason probably is that the above formulas do not, and can not, take into account the molecular energy interchanges that must take place in the combustion of one of these complex compounds. It is in all cases advisable, no matter what the fuel may be, to make calorimetric determinations of heating value.)

In his "Kalorimetrische Untersuchungen," Slaby gives the following formula for the computation of the heating value of heavy hydrocarbons. This formula in English units reads

$$H_u=[112+18880\gamma] \text{ B.T.U. per cu.ft.}, \qquad (28)$$

in which the atom number n is replaced by γ, the weight of a standard cubic foot of the hydrocarbon under consideration. Since

$$\gamma=\frac{m}{358.17} \text{ lbs.},$$

where m is the molecular weight of the gas, we may also write

$$H_u=[112+52.8m] \text{ B.T.U. per cu.ft.} \qquad (28a)$$

This formula not only applies to the individual hydrocarbons, but, with an error which does not exceed from .5 to 1%, also to mixtures of the same (as illuminating gas), when the latter contain at least 4% of hydrocarbons, have a heating value of at least 550 B.T.U. per cu.ft., and when the value of γ is quite accurately known. Table 148 shows how close Slaby's formula approximates to true results.

5. Distillation and Gasification. The conversion of the solid into gaseous fuels is done either in retorts or in special furnaces, called generators or producers.[1] In the first case the fuel is subject to distillation only, those of the hydrocarbons which can be vaporized being driven off together with the water at temperatures ranging from 900 to 1100° F., the fixed carbon remaining behind. This method (dry distillation) is the basis of the manufacture of illuminating gas, see Part IV, p. 514. In the discussion following, this gas need be no further considered since the consumer obtains it as a finished product from the manufacturer. On the other hand the construction and use of suitable power gas generators belongs strictly to the province of the gas-engine builder.

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[1] Concerning the construction of gas-making apparatus, see p. 269, Part II.

TABLE 148

HEATING VALUE OF VARIOUS HYDROCARBONS

Name of Gas	Weight in Pounds per Cubic Foot at 32° F. and 14.7 lbs. γ	Density, Air = 1.0. δ	Heating Value per Pound, Including Water of of Combustion, B.T.U. H_0	Heating Value of the Water of Combustion from 1 Pound of Gas. B.T.U. H_w	Heating Value per Pound, Exclusive of Water of Combustion B.T.U. H_u	Heating Value per Cubic Foot, Exclusive of Water of Combustion B.T.U. H_u	Heating Value from Eq. (28*a*), B.T.U., H_u
Hydrogen .... H_2	.00559	.0692	61520	9824	51696	289	....
Methane............ CH_4	.04464	.5530	23842	2457	21385	952	957
Acetylene .... C_2H_2	.07251	.8982	21429	756	20673	1499	1487
Ethylene.......... C_2H_4	.07809	.9674	21429	1404	20025	1584	1590
Ethane............ C_2H_6	.08329	1.0367	22399	1966	20433	1682	1696
Allylene........... C_3H_4	.11157	1.3819	20992	983	20009	2238	2244
Propylene.......... C_3H_6	.11699	1.4512	21224	1404	19820	2318	2330
Propane............ C_3H_8	.12256	1.5204	21825	1786	20039	2424	2435
Butylene.......... C_4H_8	.15599	1.9349	20912	1404	19508	3055	3069

The manufacture of power gas in generators also commences with a distillation of the solid fuels, but it soon changes from this directly into a gasification process which consumes the fixed carbon of the fuel, and all of the fuel is converted into gas with the exception of some incombustible ingredients (ash, clinker, etc.). In the combustion processes so far discussed, the combustion was assumed complete, that is, the resulting gas contained no combustible constituents. On the other hand, the gas made in gas producers must be a combustible gas, which first makes it necessary by some means or other (proper height of fuel column, right temperature, regulating the quantity of air introduced) to render the combustion process incomplete, that is, for instance, to burn the carbon to carbon monoxide instead of to carbon dioxide. If only carbon and air are concerned in the gasification process, the resulting **air gas**[1] can theoretically contain only carbon monoxide and the nitrogen of the air. According to p. 586, 1 lb. of carbon burned to CO_2 develops 14544 B.T.U., but only 4446 B.T.U. when burned to CO. The carbon monoxide gas produced therefore contains theoretically $14544-4446=10098$ B.T.U. per lb. of carbon gasified. This amounts to 69.5% of the heating value of the combustion of the solid C to CO_2. The remaining 30.5% are to be found in the sensible heat of the gas as it leaves the producer, and is lost if the gas must be cooled before utilization. In this case the maximum possible theoretical efficiency of the process is therefore 69.5%. It will be shown below that other methods of gasification show higher efficiencies than this, for which reason air gas is used in engines only when it can be had at a very little cost as a by-product from other manufacturing processes (blast-furnace gas, for instance).

The chemical equations of the air gas process, according to p. 585, may be written as follows:

$$\begin{array}{llll} C+O_2=CO_2+96.96 \text{ cal. per gr. mol.} & =14544 & \text{B.T.U. per lb. of} & C \\ C+O\ =CO\ +29.56 \quad \text{``} \qquad \text{``} & =\ 4446 & \text{``} \qquad\qquad \text{``} & C \\ \hline CO+O=CO_2+67.40 \quad \text{``} \qquad \text{``} & =10098 & \text{``} \qquad\qquad \text{``} & C \end{array}$$

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[1] Other names, such as "generator gas," have been used for this gas, but the term "air gas" is most suitable because it draws the distinction sharply between the carbon monoxide gas on the one hand and producer gas proper and water gas on the other, which are also made in generators or producers.

The combustion of 1 lb. of C to CO, according to p. 585, requires 1.33 lbs. of O, according to which 1 lb. of CO gas will develop during combustion $10098 \div (1+1.33) = 4334$ B.T.U. The specific weight γ of the gas is, from Table 146, .0781 lb., hence 1 cu.ft. of the theoretical air gas has a heating value of $.0781 \times 4334 = 338$ B.T.U.

Now, in reality, the air gas process is not carried through with oxygen but with air, the gasification of 1 lb. of C requiring in theory $\frac{1.33}{.235} = 5.67$ lbs. of air. The total weight of gas produced per pound of C will then be 6.67 lbs. The amount of nitrogen present is $5.67 - 1.33 = 4.34$ lbs., which is diffused through the 2.33 lbs. of CO formed during the combustion. Hence for every pound of CO there will be $\frac{4.34}{2.33} = 1.86$ lbs. of N, and 1 lb. of air gas will then consist of $\frac{1}{1.0+1.86} = .35$ lb. $= 35\%$ of combustible carbon monoxide and $\frac{1.86}{1.0+1.86} = .65$ lb. $= 65\%$ of non-combustible nitrogen. Since the densities of N and CO are very nearly alike, the same percentages express also the volume composition. The heating value of 1 lb. of this gas, since it contains 35% of CO, will be, $.35 \times 4334 = 1516$ B.T.U. One cubic foot of air gas will weigh $.35 \times .0781 + .65 \times .0783 = .07823$ lb.; 1 lb. of C consequently produces $\frac{6.67}{.07823} = 85.3$ cu.ft. of this gas.

In the production of **"water gas"** the sensible heat of the gas, which is lost in the production of air gas, is largely saved by converting it into latent heat, that is, utilizing it in the production of a second combustible gas. For this purpose the coal or coke column in a water gas generator is alternately blown into incandescence by means of air and then cooled down by passing steam through it. During the "blow" the carbon in the charge is by means of the oxygen in the air burned very largely to CO, and the temperature is constantly increasing; during the "make," water vapor replaces air and the former, in passing through the incandescent carbon, splits up into hydrogen and oxygen. The oxygen set free combines with C to form CO_2, but the latter, when the temperature is sufficiently high (above 1260° F.) is almost immediately reduced to CO. The reducing process consumes a large part of the generator heat available at the start of the make, and the temperature constantly falls. This heat is then returned by a new blowing period, bringing the temperature again up to the maximum. The water gas process is therefore carried on with constant interruptions, the blowing periods usually taking much more time than the period of gas making proper.

The higher the generator temperature at which the process is carried on, the greater the content of CO, and the smaller that of CO_2 in the resulting gas. Below from 1000 to 1250° F. the decomposition of H_2O with C takes place mainly according to the formula

$$C + 2H_2O = CO_2 + 2H_2,$$

but as the temperature increases, CO commences to outweigh CO_2, and above 1800° F. the only reaction occurring is probably

$$C + H_2O = CO + H_2.$$

According to the last equation, a given volume of the theoretically perfect (synthetic) water gas therefore consists half of carbon monoxide and half of hydrogen. The weight ratio may be determined from the molecular weights as follows:

$$1 \text{ lb. of } C+\frac{18}{12} \text{ lbs. of } H_2O=2.333 \text{ lbs. of } CO+.166 \text{ lb. of } H_2,$$

or $$1 \text{ lb. of water gas contains } \frac{.166}{2.333+.166}=.0665 \text{ lb. of hydrogen},$$

and $$\frac{2.333}{2.333+.166}=.9335 \text{ lb. of carbon monoxide}.$$

The most unfavorable condition that can exist at the end of a gas-making period is defined by the following equations:

$$1 \text{ lb. of } C+\frac{36}{12} \text{ lb. of } H_2O=3.666 \text{ lbs. of } CO_2+.333 \text{ lbs. of } H_2,$$

or $$1 \text{ lb. of water gas contains } \frac{.333}{3.666+.333}=.0825 \text{ lb. of hydrogen},$$

and $$\frac{3.666}{3.666+.333}=.9175 \text{ lb. of carbon dioxide}.$$

The amount of heat rendered latent during the decomposition process may be found from

$$C+H_2O=CO+H_2-38.78 \text{ cal. per gr. mol.}=5800 \text{ B.T.U. per lb. of } C,$$

in which it is for the present assumed that the water enters the generator already in the form of steam. The theoretically perfect water gas has the following heating value:

Due to CO	2.333× 4334=10110	B.T.U.
Due to H_2........................	.166×61 520=10212	"
Total	20322	"
Heat supplied during blowing period (rendered latent)	5800	"
Difference = heat in 1 lb. of C=	14522	"

The latter item should have been exactly the heating value of C (14 544 B.T.U.), since the process is theoretically without loss. The discrepancy, however, amounts to less than $\frac{2}{10}$ of 1%, and is probably due to various approximations in the computation.

In practical operation, the efficiency obtained in the water gas generator is only from 50 to 60%, the remainder is lost in the sensible heat carried away by the gases during the blow, through incomplete combustion, radiation, etc. Efficient generators of moderate size produce about 32 cu.ft. of gas per lb. of C (coke) gasified, and from about 24 to 29 cu.ft. per lb. of good coal.

The generation of 1 cu.ft. of water gas consequently requires from .042 to .035 lb. of coal. The amount of water used is theoretically about .025 lb. per cu.ft. of gas;

in practice, however, the amount used is considerably greater. The average composition of the commercial water gas is in volume per cent as follows:

$$50\%\ H,\ 40\%\ CO,\ 5\%\ CO_2\ \text{and}\ 5\%\ N;\ H_u \sim 280\ \text{B.T.U. per cu.ft.}$$

Weight per cu.ft., $\gamma = .044$; theoretical air required for combustion is 2.45 cu.ft. per cu.ft.

The *Delwick-Fleischer* method of making water gas differs from the common method in that during the blowing-up period the gas mad is largely carbon dioxide, not carbon monoxide. As a consequence the carbon is burned completely and a greater quantity of heat is stored up for the decomposition of water. This permits of a longer gas making period, and the process is less often interrupted.

The method of making **"power or producer gas"**[1] is a combination of the air gas and water gas processes. As the name indicates, the gas is mostly used for the production of power in gas engines. It is made by sending a certain quantity of water vapor along with the air through the incandescent column of coal or coke in the generator, in which case the oxidizing and reducing actions above discussed, between carbon and oxygen on the one hand and between steam and carbon on the other, take place practically at the same time. The final product of this process must theoretically again consist of carbon monoxide, hydrogen, and nitrogen. The introduction into the generator of air and steam combined possesses two main advantages. In the first place it serves to bind very largely (render latent) the sensible heat of the combustion products, thus raising the efficiency of the generator, and, second, permits of uninterrupted operation of the producer. The air, being admitted along with the steam, supports the combustion to such a degree, that is, keeps the temperature of the producer at such a point that the heat consumed in the decomposition of the water vapor is immediately replaced.

The individual actions and reactions developed above for air gas and water gas of course apply directly to the manufacture of producer gas. In order to avoid repetition, however, Table 149, published by H. Gerdes,[2] is given. This table shows in concise form all of the fundamental chemical relations concerned in the process of making producer gas, and by this arrangement gives a clear insight into what takes place.[3]

In actual practice the composition of the power gas is somewhat different from the values developed in the table, mainly because the theoretical ratio between C and H_2O is not maintained, a part of the C entering into other combinations. In the best practice there are obtained about 75 cu.ft. of power gas having a heating value of 146 B.T.U. per cu.ft. from 1 lb. of coal, the heating value of which is about 14 000 B.T.U. This corresponds to a generator efficiency of about 78 to 80%. By means of recovering or more efficiently utilizing the sensible heat of the gases, which at present very largely accounts for the 20% lost, this generator efficiency may be raised by from 5 to 10%.

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[1] This gas also bears the name "Dowson" gas, but with little justice, since the English engineer, Dowson, merely developed the first successful scrubbers for the gas, the method of production for which was well known before that time.

[2] Glaser's Annalen, 1902, No. 590.

[3] A more detailed exposition of the theory of the production of power gas is given by Prof. E. Meyer in the Z. d. V. D. I., 1895, p. 1524. This investigation of the process is worthy of note for its theoretical treatment.

TABLE 149

FUNDAMENTAL CHEMICAL RELATIONS CONCERNED IN THE MAKING OF PRODUCER GAS

	Weight Relations.	Total Lbs.	Volume Relations, Cubic Feet of Gas Concerned in the Reaction.					
			H_2	CO	CO_2	N_2	Total	
1. Water gas	$C + H_2O = CO + H_2$ $12 + 18 = 28 + 2 =$ $\underbrace{+51840 - 103680}_{-51840 \text{ B.T.U.}}$	30	358	358	..	..	716	Hence 30 lbs. of C produce 2576 cu.ft. of producer gas consisting of:
2. Air gas	$C + O + N = CO + N$ $12 + 16 + 16 \times 3.24 = 28 + 51.8 =$ $+51840$ B.T.U.	79.8	..	358	..	661	1019	13.9% of H_2 27.8% of CO 6.9% of CO_2 51.4% of N_2
1 and 2 together.	24 lbs. of C + 18 lbs. of H_2O + 67.8 lbs. of air = = 840 cu.ft of air	109.8	358	716	..	661	1736	1 lb. of C therefore produces 85.9 cu.ft. of producer gas of an average heating value 135 B.T.U. per cu.ft.
3. To replace heat losses (assumed)	$\frac{1}{2}C + O + N = \frac{1}{2}CO_2 + N$ $6 + 16 + 16 \times 3.24 = 22 + 51.8 =$	72.8	..	..	179	661	840	
Sum	30 lbs. of C + 18 lbs. of H_2O + 135.6 lbs. of air = = 1680.3 cu.ft. of air	182.6	358	716	179	1322	2576	

The fuel used in practice is not pure carbon, but the latter is contaminated with earthy admixtures, sulphur, etc. In making computations on any gasification process, this fact must therefore be allowed for either by using the correct C-percentage in the fuel at the outset, or by multiplying the results obtained for pure C by a factor which expresses the percentage of pure C in the fuel used. The impurities in the fuel also have a certain effect upon the composition of the gas, increasing the proportion of incombustible gases. Among the latter will always be found several per cent of carbon dioxide (see p. 520), produced mainly owing to temperature variation in the producer (see p. 600).

The average composition, volume-per-cent, of commercial power gas is about 23% CO, 18% H, 1% CH_4, 6% CO_2, 52% N. The lower heating value of the gas is about $H_n \sim 140$ B.T.U. per cu.ft., the weight per cu.ft. $\gamma = .0789$ lb., while the air theoretically required for combustion is 1.1 cu.ft. per cu.ft.

The ratio of the heating values of the individual gases in a cubic foot of the average producer gas above given to the total heating value of the cu.ft. is as follows, from Table 147.

$$\text{For CO, heating value} = .23 \times 338 = 77.8 \text{ B.T.U.} = 56.5\% \text{ of } H_u$$

$$\text{For H, heating value} = .18 \times 289 = 52.0 \text{ B.T.U.} = 37.0\% \text{ of } H_u$$

$$\text{For } CH_4\text{, heating value} = .01 \times 952 = 9.5 \text{ B.T.U.} = 6.5\% \text{ of } H_u.$$

It is evident that the hydrogen content has an important influence upon the heating value of producer gas. In the case of gas made from hard coal especially, the hydrogen content often accounts for more than one-half of the total heating value (see Table 124, p. 520). It is consequently of advantage thermally to introduce considerable water along with the air, as has already been pointed out on pages 269 and 520, as long as the percentage of H in the gas is not high enough to cause trouble through pre-ignition. A second reason for limiting the amount of water used is found in the producer itself in that, since the decomposition of water requires heat, there is a tendency to lower the temperature of the generator, which favors the formation of the undesirable carbon dioxide (see p. 600). In general, therefore, we find that the higher the percentage of hydrogen in a producer gas, the higher also that of carbon dioxide.

The most efficient hydrogen content is found between 20 and 25%, but it is also found that with this percentage only few engines run smoothly. In most engines violent explosions and knocking in the crank mechanism usually compel a reduction in the water supplied to the generator long before the most efficient percentage of H in the gas is reached. The upper limit of water used by the producer is thus evidently set in most cases by the amount of hydrogen the engine can stand under full load.

Concerning the quantity of impurities, such as tar, carried by producer gas, a point of prime importance in the operation of engines, it is difficult to give any numerical data. Only a direct examination of coal or gas can give any reliable information regarding the subject.

The best German hard coals (Rhenish, having on the average 14 400 B.T.U. per lb.) according to a number of analyses, form from .12 to .15% of tar, that is from .0012 to .0015 lb. per lb. of coal. Assuming that the gas yield per lb. of coal averages 75 cu.ft., this would correspond to a tar content of from .016 to .020 lb. of tar per 1000 cu.ft. of raw gas. Thorough washing reduces this to from .003 to .006 lb. of tar per 1000 cu.ft., and careful wet and dry purification (in sawdust purifiers) may bring it down to .0003 lb. or even less per 1000 cu.ft.

The raw gas from bituminous coals is of course much more impure, that is, it carries more tar. Thus, for instance, a sample of gas (composition in volume-per-cent, $CO_2=1.0$, $C_nH_n=.4$, $O_2=0$, $CO=31.5$, $CH_4=2.5$, $H=11.8$, $N=52.8$, heating value$=176$ B.T.U. per cu.ft.), made from an average sample of soft coal ($H\sim 12\,600$ B.T.U.) contained in the raw state from 1.27 to 1.45 lbs. of tar per 1000 cu.ft. of gas. After a thorough scrubbing in a special washer this was reduced to .006 lb. per 1000 cu.ft., which about corresponds to what is found in washed anthracite gas and is yet permissible from the standpoint of practical operation.

6. Thermo-Chemical Diagrams Illustrating the Various Gasification Phenomena.[1] Figs. 704*a–d* illustrate the thermo-chemistry of the simple combustion of carbon in air and in water vapor. The fundamental equations for the combustion of 1 lb. of C in air are stated as follows:

$$C_2+2(O+aN)_2=2CO_2+2aN_2, \quad \ldots \ldots \quad (1)$$

$$C_2+\ (O+aN)_2=2CO+\ aN_2. \quad \ldots \ldots \quad (2)$$

Eq. (1), utilizing the oxygen from 141 cu.ft. of air develops an available amount of heat equal to 14 544 B.T.U., while in eq. (2) the oxygen from only 70.5 cu.ft. of air is used, resulting in a development of heat equivalent to 4446 B.T.U., the remaining 10 098 B.T.U. being chemically bound in the CO produced. Between these two limits, of course, any combination of these two types of combustion is possible. Attention is called at this point to the latent heat of carbon which must be expended in converting the solid C to the gaseous form. The fact that heat must be so expended is the reason why the heating value of C is not 20 196 B.T.U. per lb., but only 14 544, and why the heat developed from the combustion of C to CO is not exactly half that of the combustion of C to CO_2.

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[1] Karl Kutzbach. Taken from an article published in the Z. d. V. D. I., 1905, p. 233, with the permission of the author.

In Fig. 704*a*, Diagram 1 shows the volume relations between the gases resulting from the combustion of unit weight of carbon according to eqs. (1) and (2). This diagram is then transposed into Diagram 2, Fig. 704*b*, showing the volume relations based on the cubic foot, and Diagram 3, Fig. 704*c*, showing the weight relations based on the pound. Diagram 7, Fig. 704*d*, shows the thermal relations existing during both combinations.

Diagram 1 is obtained as follows:

Combustion according to eq. (1):

$$C_2+2(O+aN)_2=2CO_2+2aN_2. \qquad (1)$$

Weight relation:

$$24+64+56a=88+56a$$

or

$$1\text{ lb. C}+2.66\text{ lbs. O}+2.33a\text{ lbs. N}=3.66\text{ lbs. CO}_2+2.33a\text{ lbs. N.}$$

Since the oxygen in this combustion is obtained from air, and the weight relation between N and O in air $=\frac{76.5}{23.5}$, the value of a in the above equation must be

$$\frac{76.5}{23.5}\times 2.66=2.33a,$$

$$a=3.73.$$

Substituting this value of a in the above equation, we finally have

$$1\text{ lb. C}+2.66\text{ lbs. O}+8.70\text{ lbs. N}=3.66\text{ lbs. CO}_2+8.70\text{ lbs. N.} \qquad (1a)$$

Now to change eq. (1*a*) into a volume relation we simply employ the specific weight γ for each gas from Table 142. The specific weight of gaseous carbon is taken at .0668 lb. per cu.ft. We have for 1 lb. of C,

Volume Relation :

$$14.97\text{ cu.ft. C}+\underbrace{29.93\text{ cu.ft. O}+111.1\text{ cu.ft. N}}_{141\text{ cu.ft of air}}=29.93\text{ cu.ft. CO}_2+111.1\text{ cu.ft. N.} \qquad (1b)$$

Combustion, according to eq. (2), treated in a similar manner, gives the following relations:

$$C_2+(O+aN)_2=2CO+aN_2. \qquad (2)$$

Weight relation:

$$24+32+28a=56+28a,$$

or

$$1\text{ lb. C}+1.33\text{ lbs. O}+1.17a\text{ lbs. N}=2.33\text{ lbs. CO}+1.17a\text{ lbs. N.}$$

Putting $a=3.73$ as before, we have

$$1\text{ lb. C}+1.33\text{ lbs. O}+4.35\text{ lbs. N}=2.33\text{ lbs. CO}+4.35\text{ lbs. N.} \qquad (2a)$$

Volume relation. This is obtained as above.

$$14.97\text{ cu.ft. C}+\underbrace{14.97\text{ cu.ft. O}+55.55\text{ cu.ft. N}}_{70.5\text{ cu.ft. of air}}=29.93\text{ cu.ft. CO}+55.55\text{ cu.ft. N.} \qquad (2b)$$

With the aid of eqs. (1*b*) and (2*b*) the construction of Diagram 1, Fig. 704*a*, becomes obvious. The ordinate marked 2 (in a circle) represents the relation of volumes at the end of the combustion according to eq. (2), that is, after 70.5 cu.ft. of air have been supplied. The original volume of 1 lb. of gaseous C, 14.97 cu.ft., has at this point been reduced to zero, and 29.93 cu.ft. of CO have appeared in its stead. At the same time 55.55 cu.ft. of N were supplied along with the O, making the total cubic feet of gas 85.48. Now as the supply of air is increased beyond 70.5 cu.ft., CO_2 commences to make its appearance at the expense of CO, until, when the supply of air has reached 141.03 cu.ft., all of the CO has disappeared, while 29.93 cu.ft. of CO_2 appear in its place. The cubic feet of N supplied in the meantime has increased to 111.10, making the total volume of gas along the ordinate marked 1 (in circle), indicating the end of combustion according to eq. (1), equal to 141.03 cu.ft. For any air supply beween 70.5 and 141.03, both CO and CO_2 appear in the diagram, the relation between them always being

$$\text{Volume CO} + \text{volume } CO_2 = 29.93 \text{ cu.ft.}$$

The conventions used in this diagram, together with some other data relating to some of the other diagrams, are shown in Fig. 703.

Convention Used.	Wt. in Lbs. per Cu. Ft.	Heating Valve per Cu. Ft.	Cu. Ft. per Lb.	Heating Value per Lb.
C ▦ 12	—	—	14.97	14500
O ▥ 16	.08921	—	10.22	—
CO ▥ 28	.07807	342	12.81	4320
CO_2 ▥ 44	.12267	—	8.17	—
N ▨ 14	.07831	—	12.81	—
Air 21 Vol % O+N ▨ 23.2 Wt %	.08072	—	12.51	—
H ▤ 2	.00559	Higher 346 297 Lower	179.7	Higher 61560 51730 Lower
H_2O ☐ 18	.05016	—	19.4	—

FIG. 703.—Key to Figs. 704 and 705.

Diagram 2, Fig. 704*b*, is obtained from Diagram 1 by merely expressing the volume relations existing at ordinates (1) and (2) in percentages of the total volume, while Diagram 3, Fig. 704*c*, is constructed by aid of eqs. (1*a*) and (2*a*). In Diagram 2 the curve of heating value has been obtained by determining the heating value of .347 cu. ft. of CO. This is equal to 117 B.T.U. Now as more air is supplied some of the CO burns to CO_2, and the heating value of each cubic foot of the gas is less as we go beyond 70.5 cu.ft. of air, until at ordinate (1) all of the CO has been burned to CO_2, and the heating value is then reduced to zero, all of the gas being incombustible. The curve is plotted to the same scale as the scale of percentage at the side of the diagram.

If steam be led through incandescent carbon, the resulting gas consists of a mixture of H, CO, and CO_2. The reaction proceeds according to either or both of the following equations:

$$C_2 + 2H_2O = 2CO + 2H_2, \quad \ldots \quad (3)$$

$$C_2 + 4H_2O = 2CO_2 + 4H_2. \quad \ldots \quad (4)$$

Determining the volume and weight relations represented by these equations, as before, we have

From eq. (3):

Weight relation: $24 + 36 = 56 + 4,$

or

$$1 \text{ lb. C} + 1.5 \text{ lbs. } H_2O = 2.33 \text{ lbs. CO} + .166 \text{ lb. H.} \quad \ldots \quad (3a)$$

This equation is converted to the volume relation by the aid of the specific weight γ from Table 146.

Volume relation: 14.97 cu.ft. of $C+29.93$ cu.ft. $H_2O=29.93$ cu.ft. $CO+29.93$ cu.ft. H. . (3*b*)

From eq. (4):
Weight relation: $24+72=88+8$,

or 1 lb. $C+3$ lbs. $H_2O=3.66$ lbs. $CO_2+.333$ lb. H. . . . . (4*a*)

For this we obtain, as before,

Volume relation: 14.97 cu.ft. $C+59.88$ cu.ft. $H_2O=29.93$ cu.ft. $CO_2+59.88$ cu.ft. H. . . . (4*b*)

Equations (3*a*), (3*b*), (4*a*), and (4*b*) serve to plot Diagrams 4, 5, and 6, Figs. 704 *a* to *c*. The method of doing this is similar to that used for Diagrams 1, 2, and 3, and need hardly be outlined any further. The heating value curve in Diagram 5, Fig. 704*b*, is obtained by determining the heating value of .667 cu.ft. of H at ordinate (4), and the heating value of .5 cu.ft. of $CO+.5$ cu.ft. of H at ordinate (3). This determines the end-point of a curve for which intermediate points may be found from the volume relations in the same diagram.

Finally, Diagrams 7 and 8 in Fig. 704*d* show the various quantities of heat concerned in the reactions for both air gas and water gas. The construction of these diagrams is obvious, the various quantities of heat being computed directly from the weights involved in eqs. (1*a*), (2*a*), (3*a*), and (4*a*).

All of the diagrams between the end conditions at ordinates (1) and (2), or (3) and (4), show interactions resulting in the formation of both CO and CO_2, and the question next comes up as to what influence determines the relative formation of these two gases. This controlling factor is the temperature. There is a chemical law which states that with increasing temperature the chemical actions and interactions are so modified as to resist the further increase of temperature, that is, that those reactions which take place with a smaller development of heat or which are combined with a cooling process gain the ascendency over other possible reactions. In the air gas process, the reaction combined with the smaller development of heat is evidently that according to eq. (2), that is, the formation of CO. In the water-gas process the reaction showing the greatest cooling effect is that according to eq. (3), again the formation of CO, because as Diagram 8, Fig. 704*d*, shows, at ordinate (3) the latent heat in the gases is 18 679 B.T.U., while at ordinate (4) it is only 17 244 B.T.U.

In actual practice the temperature is of course by no means the same in all parts of the producer. Under normal conditions, it probably increases from the grate upward until all of the oxygen in the air has been combined, after which it decreases on account of the heat furnished to the green coal moving downward. In any part of the producer the relative formation of CO and CO_2 depends entirely upon the temperature, the transformation from the CO_2 formed to CO in the hotter zones going on at the same time that some CO is again converted into CO_2 in the cooler zones. The composition of the producer gas as it finally leaves the generator is a compromise between these combinations and dissociations at different temperatures, but the net result is that the highest temperature, on account of its greater potency, is the deciding factor.

Examining again the heat diagrams for air gas and water gas, it will be noticed that the former shows a considerable quantity of sensible heat, which as far as the obtaining of the greatest possible amount of chemical energy is concerned, is wasted. The heat diagram for the water-gas process on the other hand shows a deficiency of heat, since C itself can only furnish 14 544 B.T.U. per lb. At ordinate (3) there is a deficiency of 4176 B.T.U.; at ordinate (4) of 2700 B.T.U. Both quantities referred to steam at 32° F. If now *both* steam and air are introduced together, the sensible (waste) heat of the air-gas process can supply or balance the deficiency of the water-gas process.

The diagrams of Figs. 705*a* to *c* illustrate this combined air-steam (producer) gas process. Diagrams 9, 10, and 11 refer to the perfect gasification process, the efficiency being assumed 100%.

Diagram 9, Fig. 705*a*, showing the "heat balance" for the perfect process, indicates the method of finding the balance or state of equilibrium above mentioned. The three-coördinate system is used, the three coördinates representing the variables: air, steam, and heat units, as indicated. The diagram in the left vertical plane is a reproduction of Diagram 7, Fig. 704*d*; that in the right vertical plane one of Diagram 8. With their aid it is possible to determine the conditions at which the waste heat of the air-gas process just balances the deficiency of the water-gas

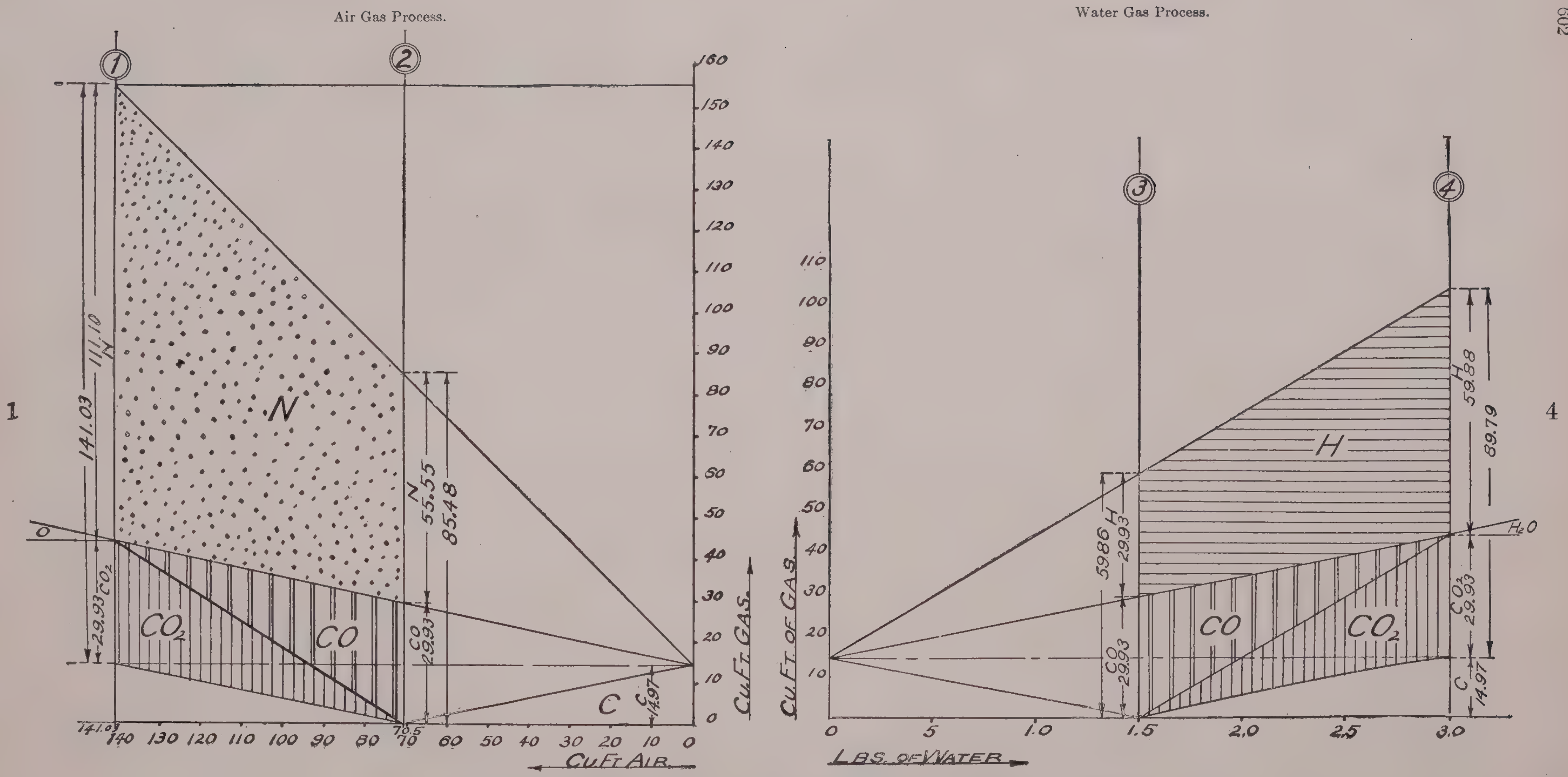
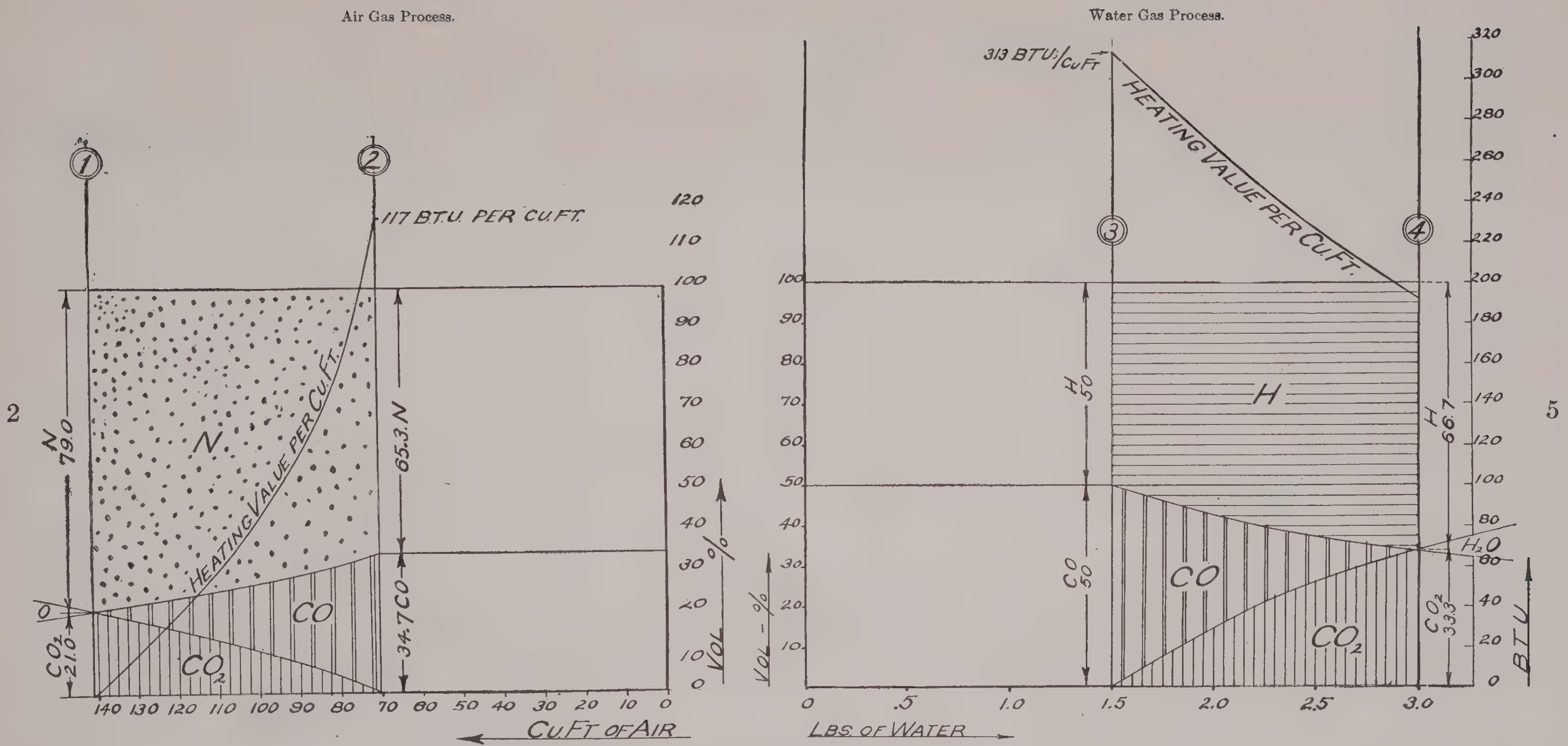


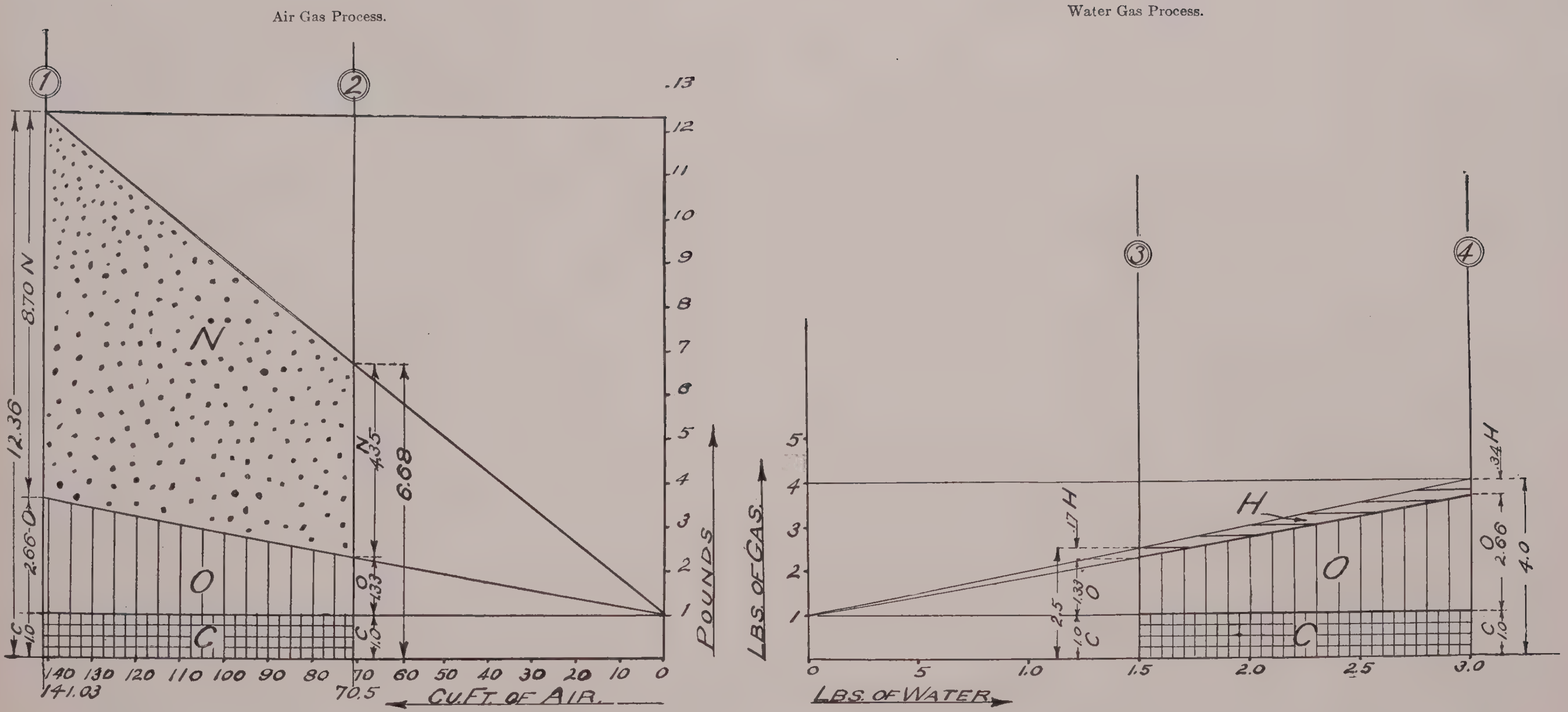
FIG. 704*a*.—Volume Diagrams per Pound of C gasified, Air Gas and Producer Gas Processes.



Air Gas Process.

Water Gas Process.

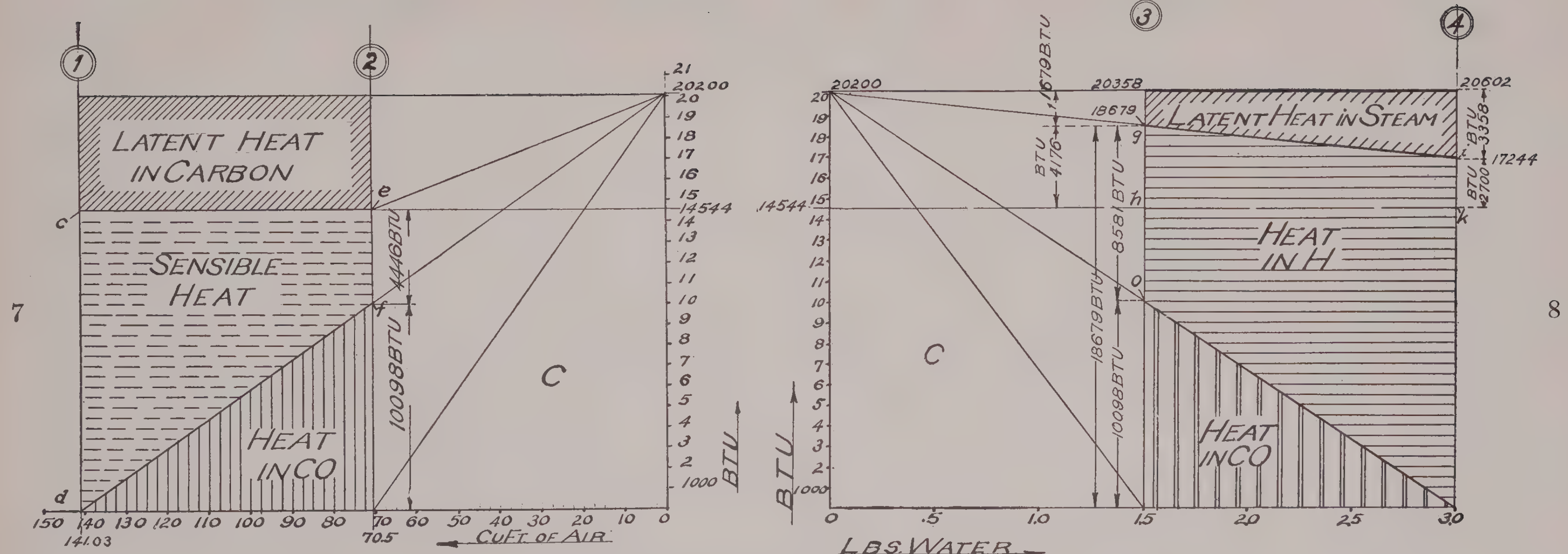
FIG. 704*b*.—Volume Diagrams for One Cubic Foot of Gas, Air Gas and Producer Gas Processes.



Air Gas Process.

Water Gas Process.

FIG. 704c.—Weight Diagrams for Air and Water Gas Processes.



Air Gas Process.

Water Gas Process.

FIG. 704*d*.—Heat Diagrams for Air and Water Gas Processes.

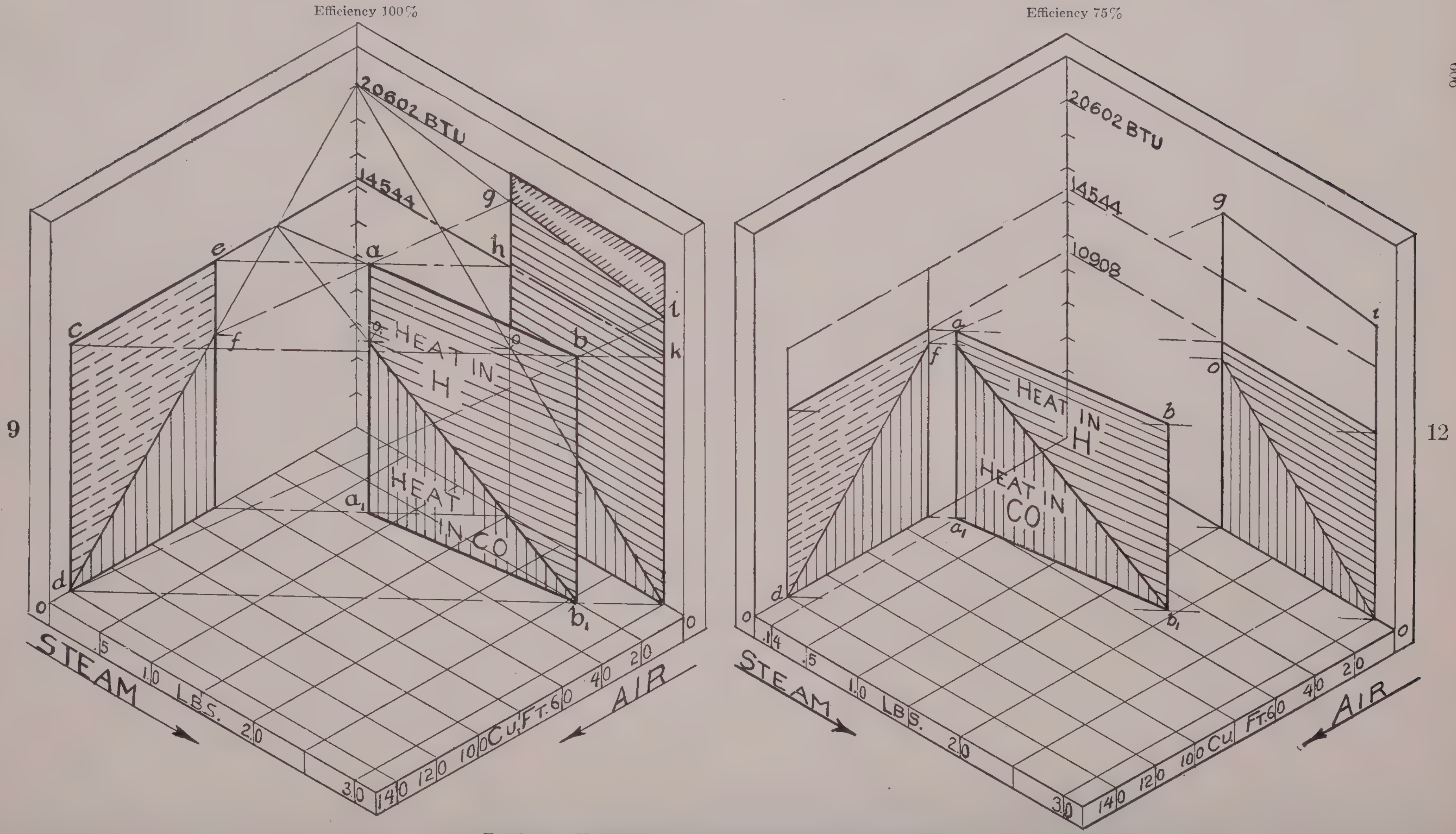
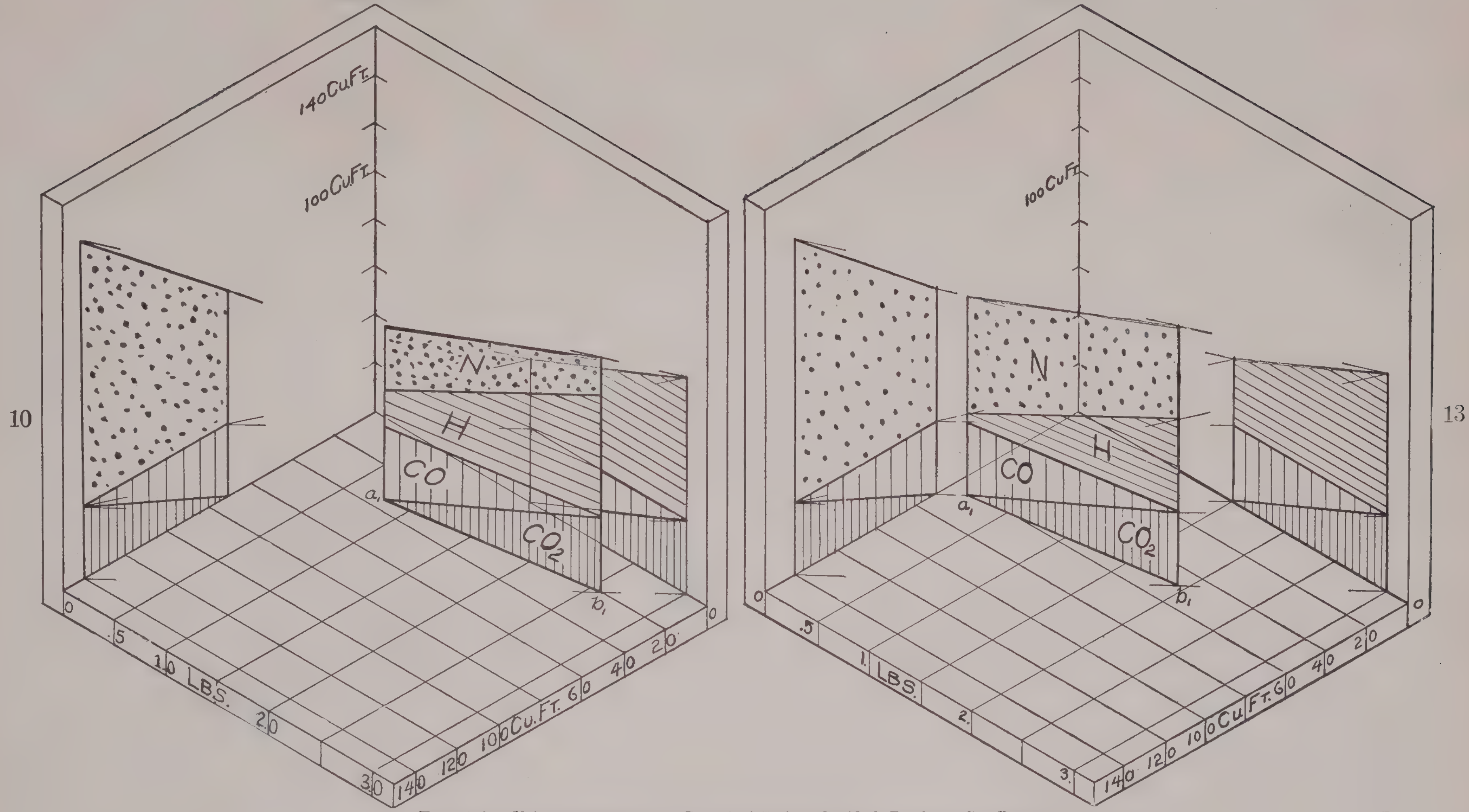


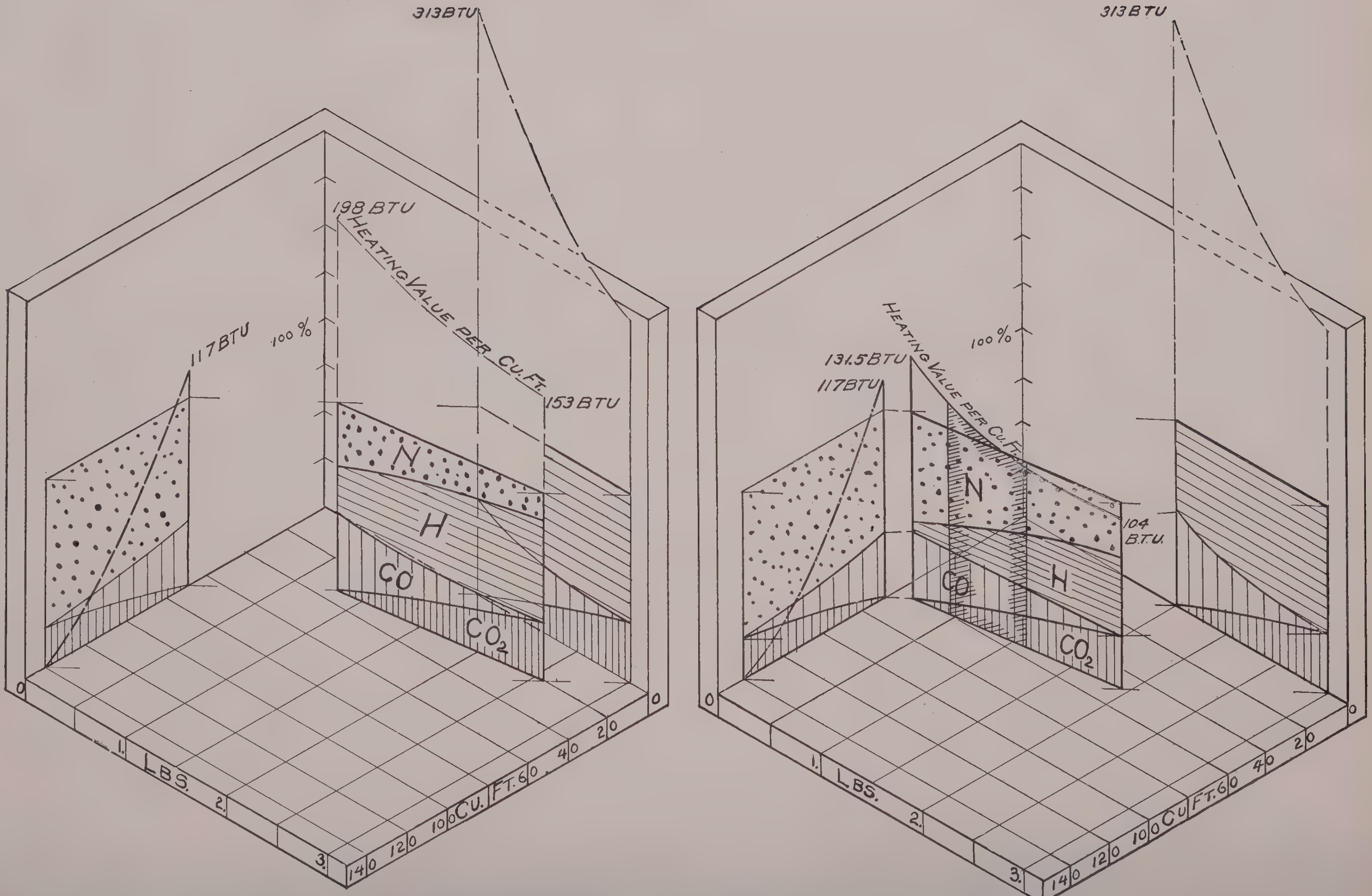
Fig. 705*a*.—Heat Balance Diagrams for Producer Gas Process.



Efficiency 100%.

Efficiency 75%

Fig. 705*b*.—Volume Diagrams per Pound of Carbon Gasified, Producer Gas Process.



Efficiency 100%

Efficiency 75%

FIG. 705c.—Volume Diagrams for One Cubic Foot of Gas, Producer Gas Process.

process. Points *c, d, e,* and *f* defining the waste heat of the air-gas process, are connected as shown with points *k, i, h,* and *g,* defining the heat deficiency of the water-gas process. The points of intersection *a* and *b* and their vertical projection a_1 and b_1, defining the producer-gas diagram, allow of the determination, in the horizontal plane, of the relative quantities of air and water.

The diagram shows that at the point a_1 for instance, the quantities to maintain a balance furnished per pound of C should be .77 lb. of steam and 34.5 cu.ft. of air. At this point there will be no CO_2 found in the gas. At the point b_1 the respective quantities per pound of C are 22.46 cu.ft. of air and 2.53 lbs. of steam. At this point all of the CO will have been converted to CO_2.

The upper-end point o_1 of the line defining the heat in CO in the producer-gas diagram is found by drawing the line *of.* The CO area in the producer-gas diagram is, in the nature of the case, the same as in the other two diagrams. The total sensible heat of the air-gas diagram now appears as heat in H in the producer-gas diagram.

The assumption of a generator efficiency equal to 100% presupposes that the sensible heat of the materials entering the gasification process is at least equal to the sensible heat of the gas leaving the producer. In actual practice, however, the gases leave the gasification zone of the producer at the combustion or reduction temperature. They consequently carry along a considerable stock of sensible heat of which only a small part is transferred to the coal on the way out for drying, distilling, etc., and another part is recovered for pre-heating purposes. A large part of this sensible heat is consequently lost, and the efficiency of the process is therefore at once reduced, the actual gasification efficiency ranging from 75 to possibly 85% as a maximum.

If we assume as a lower limit an efficiency of 75%, the available heat is reduced from 14 544 to 10 908 B.T.U. per lb. of C, while the waste heat of the air-gas process is only 810 B.T.U. This of course means that less heat is available for the water-gas process and that consequently less water can be evaporated. The heat balance for this process is shown in Diagram 12, Fig. 705*a*, the construction of which is obvious. The producer-gas diagram in this case shows, when no CO_2 is produced (point *a*), that to maintain a balance .14 lb. of water and 64.18 cu.ft. of air must be furnished per pound of C.

Diagrams 10 and 13, Fig. 705*b*, next show the volumes of the various gases produced per pound of carbon gasified. Their construction, once the position of what might be called the "equilibrium" line a_1b_1 is found, is easy and obvious.

Finally, Diagrams 11 and 14, Fig. 705*c*, expressing the composition of the producer gas made in volume-percent are directly obtained from Diagrams 10 and 13. These diagrams also show the curve of heating value per cubic foot of producer gas. Attention is called to the decided variation in the production of hydrogen depending upon the producer efficiency and the amount of CO_2 made.

Up to this point all of the discussion of the gasification process has been based upon the assumption that pure carbon was used. In practice, however, we find carbon available only with certain admixtures. Coke always contains less carbon than the undistilled fuel because a part of the carbon together with hydrogen forms the volatile ingredients, like tar and the light hydrocarbons of illuminating gas. Distillation commences at comparatively low temperatures, 900 to 1400° F., and is therefore likely to be complete when the gasification zone of the producer is reached, especially for small sized fuel. The only important factor in the gasification process itself is then the carbon in the coke. To obtain the composition of the gas resulting in such a case, the gases of distillation (illuminating gas) obtained for 1 lb. of C may, according to volume and composition, be drawn in above the quantities of producer gas shown in the volume diagrams (either 10 or 13) of Fig. 705*b*. Under certain circumstances this may result in a considerable increase in the heating value of the producer gas, since the heating value of illuminating gas may be from 450 to 575 B.T.U. per cu.ft. The percentage of CH_4 found in producer gas and a part of the H content is always due to the gases of distillation from the coal as fired. The total efficiency of the producer consequently always exceeds the mere gasification efficiency, and the gain is the greater the higher the percentage of volatile matter (hydrocarbons) in the coal.

In Diagram 14, Fig. 705*c*, the range of producer gases usually made in the normal operation of a producer plant, depending upon the supply of steam, is indicated by the area in the producer-gas diagram marked by marginal hatching. Of these gases the one containing most H, and consequently also most CO_2, corresponds about to the gas made by the Mond process, although the H content of this gas may, with coal high in volatile matter, be even considerably higher than that shown. The low-temperature operation of the Mond process permits of the obtaining of an excellent producer efficiency, which is, however, somewhat reduced owing to the large quantity of steam (up to three-fourths of that furnished) passing through the generator undecomposed.

B. SOME DETAILS FROM PRACTICE[1]

I. Directions for Operation, Attendance, Etc.

Instructions for the *operation* and *care* of internal-combustion engines, intended for the layman, as distinguished from the engineer or skilled attendant, can hardly be made too elementary. Such directions should clearly and concisely point out the operations that must be gone through in daily routine and should, besides this, also give quick and safe advice in the case of any troubles out of the ordinary that may arise. An expert erector will personally give such instructions to the regular operator in a very few hours, but the written word, as always, calls for distinctness and exactitude even at the expense of brevity.

There is of course little use in presenting here any sets of instructions as examples, because each different make of engine requires its own distinct set. But in order to show both the language employed and the very great help given by pictures, etc., of certain parts, there are given below instruction books and directions issued by one or two well-known American firms.

Besides giving a general key to the main engine parts, any set of instructions must consist of comprehensive and logically arranged series of rules for starting and stopping and should include hints for the hunting down of troubles and for eradicating them.

Instructions concerning *producer installations* in general read much alike, no matter what the make of the producer. The reason for this is that, although the various operations required depend somewhat on the constructive details of the producer, the latter are much less diverse than is the case in gas engines, and, fundamentally considered, all gas producers are nothing but a special type of furnace. Directions for judging of the proper state of the gasification process, for recognizing and removing unusual troubles, etc., are consequently much the same in all cases. The following pages give the directions and instructions issued in the case of three well-known makes of producer installations, mainly to serve as a guide for similar purposes.

Special emphasis should be laid on such statements or points in the instructions, the neglect of which may lead to dangerous occurrences (accidents to attendants or to machinery). In this connection it should be remembered that in some instances of this kind the manufacturers may be legally responsible to either the purchaser or the persons injured, and for that reason any instructions aiming to point out the possible serious consequences of certain wrong management or attendance should be made specially conspicuous.

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[1] Translator's Note. This division of the book does not strictly follow the German text, for the reason that some of the material there given is of no importance to American readers. Thus the directions given for the operation and care of a kerosene engine have been replaced by others issued by American makers. The instructions concerning gas producers were translated for reasons given below.

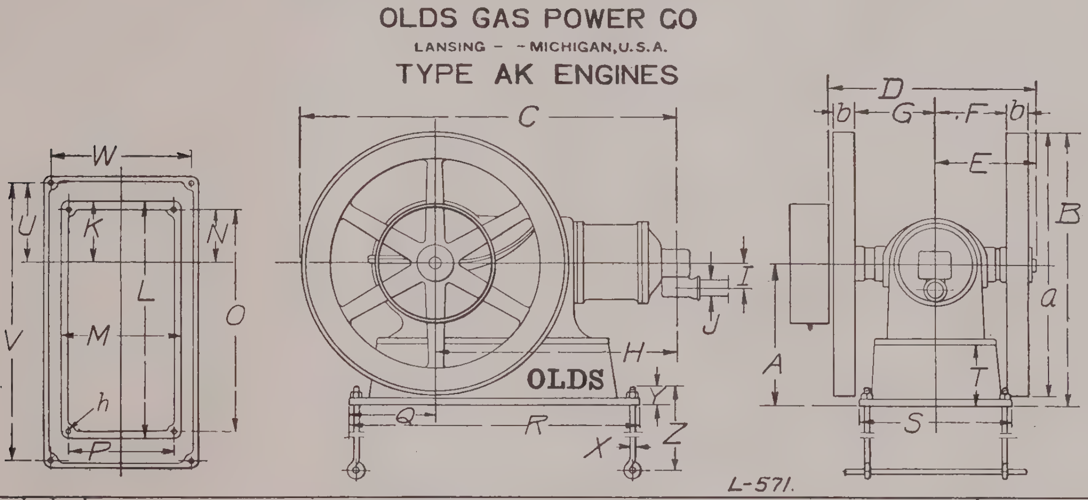
The original text contains nothing concerning the methods of testing internal-combustion engines, and on account of the importance of this subject both the Code of the American Soc. of Mech. Engrs. and that of the German Society of Engineers are inserted. It is understood that the former is at the present writing being revised by the Gas Power Section of the American Society.

1. Instruction Book for Olds Kerosene Engines, Type AK

Olds Gas Power Co., Lansing, Mich.

DON'T WORRY

Do not let the size of this book alarm you. Olds engines are simple, easy to run, easy to understand and reliable. We have made this book large simply because we wish to cover every possible point and put you in shape to take care of your engine yourself. If you study this book and understand it, you will have no trouble. If you do have trouble, whatever it is, if you read this book and understand it you can fix it yourself.



TYPE	H.P.	R.P.M.	DIMENSIONS GIVEN IN INCHES.																												PULLEYS			
			GENERAL DIMENSIONS.										ENGINE BASE							SUB BASE.							ANCHOR BOLT			WHEEL		ST'D.		SMALL	
			A	B	C	D	E	F	G	H	I	J	K	L	M	N	O	P	h	Q	R	S	T	U	V	W	X	Y	Z	a	b	DIA.	BELT	DIA.	BELT
2AK	3	550	15 1/2	29 1/2	40 1/4	21 5/8	10 5/16	6 15/16	7 13/16	23 3/4	2	1 1/4	5 3/4	22	11	5	20 1/2	9 1/2	1/2	8 3/4	28	14	8 1/2	8	26 1/2	12 1/2	1/2	2	16	28	2	6	4	—	—
3AK	4 1/2	500	17 1/2	31 1/2	44 3/4	24 1/4	11 11/16	8 9/16	8 13/16	28	2 3/8	1 1/2	6 1/4	25 1/2	13	5 1/2	24	11 1/2	1/2	9 1/2	32	16 1/2	9	8 3/4	30 1/2	15	1/2	2	16	28	2 1/4	8	5	6	4
4AK	6	450	19	35 1/2	46 1/8	26 5/16	13 1/2	9 1/8	9 9/16	26 3/4	3 1/4	2	7 1/4	27	14	6 3/8	25 1/4	12 1/4	1/2	10 3/4	34	18 1/2	10	9 7/8	32 1/4	16 3/4	5/8	2 1/2	20	33	2 5/8	10	5	—	—
5AK	8	425	20 1/2	38 1/2	50 7/8	26 11/16	13 7/8	9 9/16	9 11/16	29 13/16	4 1/4	2	6 7/8	28 3/4	14 1/4	6	27	12 1/2	5/8	10 3/4	36 1/2	18 3/4	11	9 7/8	34 3/4	17	5/8	2 1/2	20	36	3	12	6	—	—
6AK	12	400	21 1/2	41 1/2	59 7/8	33 1/4	17 1/4	10 11/16	11 1/2	35 5/8	4 7/8	2 1/2	9	35 1/2	18	8	33 1/2	16	3/4	13	43 1/2	23	9	12	41 1/2	21	3/4	2 1/2	28	40	3 1/4	18	6	—	—

Fig. 706.

Four-Cycle. The Olds engine belongs to the "four-cycle" type—that is, it gives one explosion every two revolutions of the fly-wheel. Each two revolutions of the fly-wheel complete a cycle of four operations. Look at your engine, and we will take them in their order.

1. *Charging or Suction Stroke.* The crank has pushed the piston as far back into the cylinder as it will go, and now as it goes on turning, it starts to pull the piston out again—the inlet valve opens, and as the piston moves forward, the suction pulls in a charge of mixed fuel and air until the space behind the piston is completely filled, when the inlet valve closes.

2. *Compression Stroke.* The piston has gone out its full distance, the cylinder is full of gas, and the piston now starts back in, compressing this mixture of gas and air. Just as the piston is reaching the end of its compression stroke, the electric spark ignites the charge, the

3. *Explosion or Expansion stroke* takes place, and the piston is pushed out with great force by the explosion of the gas. This is the working stroke, the energy of which is transmitted to and stored by the fly-wheels. The piston rushes out to its full limit and returns on the

4. *Exhaust Stroke.* Before the working stroke was quite completed, the exhaust valve had been already thrown open to avoid back pressure, and the returning piston now simply sweeps the cylinder of burnt gases, driving them before it and out through the exhaust valve. Your cylinder is cleaned and swept, so to speak, ready for a new charge, and your cycle is complete.

INSTRUCTION BOOK OLDS TYPE AK ENGINES

This booklet is intended to instruct explicitly in the operation of the Olds kerosene engine. It has been gone into so carefully and so thoroughly that if it is studied intelligently it will be easy for anyone to run our engines and keep them running, whether experienced or not.

Before the engine leaves our hands the adjustments have all been made by us, and it is ready to run.

Do not tamper with anything until you have fully mastered these instructions, and know just what you are doing, and why, and what the result will be. Do not turn a nut to tighten it or loosen it, or fix something that may seem loose, or loosen something that may seem tight, just out of curiosity, as this will generally make trouble for you.

Remember that it is absolutely necessary to use gas engine oil in lubricating cylinder and piston—no other kind of oil will do.

Instructions for Setting Up. Do not remove the crating until the engine has been brought to the place where it is to be used. In taking off the crating, be careful not to break anything inside. Set the engine on the foundation, and always remember that the firmer the foundation the smaller the repair bills. If a cement foundation is used there must be anchor bolts coming up through the cement base to fit the holes in the bed of the engine. If the engine is placed on the floor or on skids use lag bolts or regular bolts.

The table of dimensions (Fig. 706) gives the exact size of these, and the location of these holes.

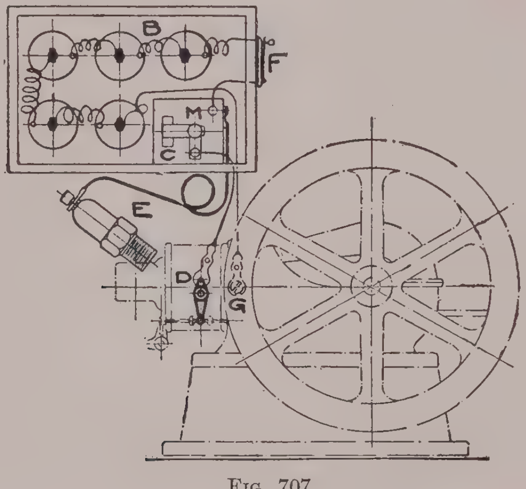


Fig. 707.

Ignition System. The ignition system is what is known as the jump spark, in which an induced current of high tension is caused to jump a gap between electrodes in a plug screwed in the cylinder head. The current for this system can be supplied by batteries, magneto, or both together. Below are shown three cuts. Fig. 707 shows the connections between the batteries, the coil, the contact maker, and the spark plug. For clearness the cut shows the batteries and plug enlarged out of proportion to the engine.

Fig. 708 shows the arrangement of the batteries and magneto where these two systems are installed together, either one to be used.

Fig. 709 shows arrangement where magneto is used alone.

Using Battery Only. In the crate you will find a large red box. This is the battery box The spark coil and the batteries are inside—and they are properly wired together—on the inside. See that in each case the connection is tight—screwing down any of the post screws that happen to be loose (one on the coil and two on each battery at the point where the wires make the contact).

The wires come through a hole in the battery box, and on the outside of this battery box is a coil of wire. This wire is to be used in connecting up the battery box with the engine.

Open the battery box with a screw driver, and you will find two spark plugs in the box; one is for actual use, and the other is for reserve.

See that the points in the spark plug are about $\frac{1}{32}$ of an inch apart. If they are too far apart the engine will skip, and if they are too near together you will not get a big enough spark to explode the charge. You can bend them slightly with a pair of pliers. Be very careful in doing this, as the spark plug is delicate.

Set the battery box near enough to the engine to have the wires reach easily. The large wire is to be fastened to the spark plug *E*, Fig. 707. If there is a terminal attached to the wire, this fits under the screw on the spark plug. Force it gently around the spark plug post and turn the thumb screw down until it holds tightly. If there is no terminal on the wire, cut away some of the insulation, or rubber covering, at the end of the wire, and take this bare wire and wrap it around the top of the spark plug post, then tighten it with pliers so that the connection is tight. Take the small wire that comes out of the box nearest to the big wire and fasten this on top of the commutator *D*, Fig. 707. The other wire coming out of the box nearest the switch is fastened to the engine bed under the round-headed screw *G*, Fig. 707. See that the short insulated wires connecting the five cells together are screwed down tight. Should any wires become disconnected, see diagram, Fig. 707, for wiring.

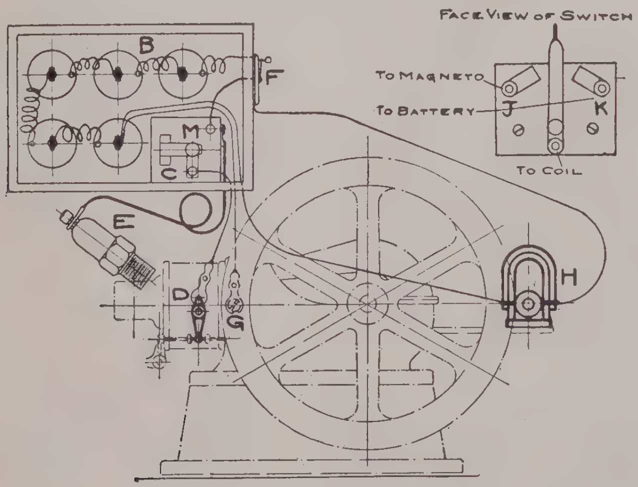


Fig. 708.

Using Batteries and Magneto. Batteries and magneto are to be arranged so that they can be run alternately at will. The engine should be started on the batteries, the batteries then thrown off and the running done on the magneto. In Fig. 708, *H* is the magneto, and *F* is the two-point switch. When the switch is on point *J* (running position) the magneto is connected in the circuit, and furnishes current to the system; and when it is on point *K* (starting position) the battery furnishes the current.

The arrangement (Fig. 709) shows the connections where the magneto alone is used.

If a magneto is used, the commutator should be kept clean with a little gasoline and waste, and any glaze which is on the rubbing surface of the brushes should be removed with a knife. The brushes are purposely made difficult to get at, and without adjustment, as once set they should remain in adjustment until worn out. A direction sheet accompanies each magneto, and goes into the subject of the magneto quite thoroughly. The magneto furnishes a direct current of low tension of from 7 to 9 volts. If it is desired to substitute batteries for magneto, or supplement the magneto with the batteries, enough batteries should be supplied to give 6 volts.

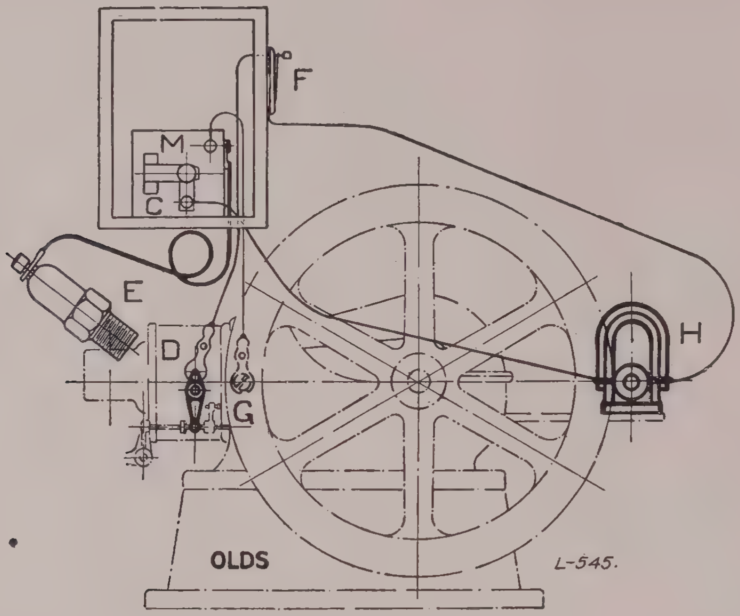


FIG. 709

The Coil. The spark coil *C*, shown in Figs. 707, 708, and 709, consists of a primary coil through which the current from the batteries or the magneto is carried, the vibrator which interrupts the circuit automatically in this primary coil, and the secondary coil in which is induced a current of high voltage, due to the interruption by the vibrator of the current in the primary coil. Further, in the primary circuit, is interposed a condenser made of tin foil and paraffin paper.

The principal point to avoid in the use of coil is dampness; if the coil becomes soaked with rain, it should be dried out very slowly at a low temperature, so as not to melt the paraffin in the condenser.

Muffler and Water Piping. Fig. 710 shows the muffler piping for all sizes; Fig. 711 shows the water piping for 2AK and 3AK engines. Fig. 712 shows same for 4AK, 5AK and 6AK. Below each cut is a little printed table giving the location and size of each pipe and fitting for each engine. The piping and fittings will be found in a box in the crate of the engine; and if careful attention is given to putting on the connections exactly as shown, they will be found to go together properly. If you have to carry the exhaust further from the muffler than the short pipe furnished will reach, the extra piping for this purpose must be of a larger size than the pipe from the muffler. In piping the exhaust outside a building, have the hole in the building through which it goes out lined with tin, or some non-conductor of heat to prevent the wood of the building being scorched. Always have the muffler as close to the engine as possible, and carried on foundation or support of some kind so the weight of the muffler will not be hanging on the flange of the valve cage. In case it is desirable to pipe the exhaust out doors, it is usually advisable to set the muffler just outside the building, provided this can be done by carrying the muffler not more than 10 ft. from the engine. The muffler should be placed in such a position that it cannot fill up with water, either from rain or any other cause. Long exhaust pipes are bad. They cause back pressure on the engine; turns in the pipes should be avoided and the exhaust should be allowed to go as directly out of the muffler as possible. If the exhaust pipe is long, condensation of moisture might drain back into the muffler and seal the muffler off, preventing starting. There is a drain on the bottom of the muffler. Open this drain occasionally and drain off any water in the muffler.

In the water pipe system drains are provided for the tank and cylinder jacket. It is advisable to clean out the tank and drain off the water from the jacket occasionally to remove any dirt or scale which may accumulate; and in cold weather the water should **always** be drained off to prevent freezing and breakage.

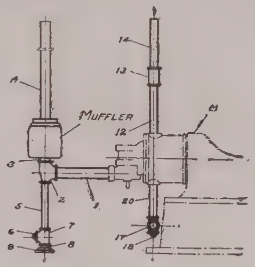


Fig. 710.

EXHAUST PIPING

2AK. 1. 1¼×12″ W. I. Pipe.
2. 1¼×1″ Reducing Tee.
3. 1¼″ Close Nipple.
4. 1¼″×6′-0″ W. I. Pipe (thread one end only).
5. 1×8½″ W. I. Pipe.
6. 1″ Pipe Plug.
7. 1″ Mall. Tee.
8. 15/16×1½″ C. R. S. with Pipe (thread on both ends).
9. 1×4″ C. I. Flange.

3AK. 1. 1½×12″ W. I. Pipe.
2. 1½×1×1½″ Reducing Tee.
3. 1½″ Close Nipple.
4. 1½″×6′-0″ W. I. Pipe (thread one end only).
5. 1×10¼″ W. I. Pipe.
6. 1″ Pipe Plug.
7. 1″ Mall. Tee.
8. 15/16×1½″ C. R. S. Pipe (thread on both ends).
9. 1×4″ Flange C. I.

EXHAUST PIPING

4AK. 1. 2×12″ W. I. Pipe
2. 2×1×2″ Reducing Tee.
3. 2″ Close Nipple.
4. 2″×6′-0″ W. I. Pipe (thread one end only).
5. 1×10¾″ W. I. Pipe.
6. 1″ Pipe Plug.
7. 1″ Mall. Tee.
8. 15/16×1½″ C. R. S. Pipe (thread on both ends).
9. 1×4″ C. I. Flange.

5AK. 1. 2×12″ W. I. Pipe.
2. 2×1×2″ Reducing Tee.
3. 2″ Close Nipple.
4. 2″×6′-0″ W. I. Pipe, (thread one end only).
5. 1×11¼″ W. I. Pipe.
6. 1″ Pipe Plug.
7. 1″ Mall. Tee.
8. 15/16×1½″ C. R. S. Pipe (thread on both ends).
9. 1×4″ C. I. Flange.

6AK. 1. 2½×14″ W. I. Pipe.
2. 2½×1½×2½″ Reducing Tee.
3. 2½″ Close Nipple.
4. 2½″×6′-0″ W. I. Pipe (thread one end only).
5. 1½×10¾″ W. I. Pipe.
6. 1½″ Pipe Plug.
7. 1½″ Mall. Tee.
8. 1⅞×2″ C. R. S. Pipe (thread on both ends).
9. 1½×5″ C. I Flange.

In case running water is used instead of a tank, the in-coming jacket water should be taken in at the lower opening of the jacket, and the overflow taken out at the top of the jacket. Where a tank is used, the circulation of water is from the bottom of the tank to the under side of the jacket, up through the jacket, out through the top of the jacket, and back to the top of the tank.

Fill the water tank with clean water so that the water in the tank is **always above the return pipe,** which is the top one. Be sure to pour this in in the morning, and keep the level in the tank always over the outlet of the top pipe. Otherwise no circulation will occur around the cylinder, and the cylinder will become overheated. Also be sure the level of the water in the tank is above the cylinder. The tanks are so built that, when the bottom of the tank is on the same level as the bottom of the engine, the pipe to the tank from the top of the engine will be high enough, and the level of the water in the tank will be above the cylinder.

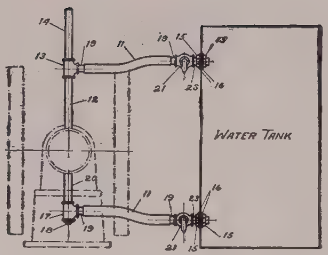


FIG. 711.

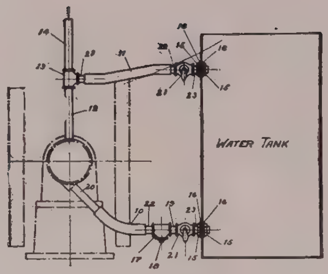


FIG. 712.

COOLING WATER PIPING

2AK. 11. $1\frac{1}{4}\times 18''$ Rubber Hose.
12. $1\times 3''$ W. I. Pipe.
13. $1''$ Mall. Tee.
14. $1\times 12''$ W. I. Pipe (thread one end only).
15. $1''$ Lock Nuts.
16. $1\frac{3}{8}\times 2''$ Rubber Washers.
17. $1''$ Mall. Tee.
18. $1''$ Pipe Plug.
19. $1\times 4''$ W. I. Pipe (thread one end only).
20. $1\times 3\frac{1}{2}''$ W. I. Pipe.
21. $1''$ Brass Cocks.
23. $1\times 3''$ W. I. Pipe.

3AK. 11. $1\frac{3}{4}\times 18''$ Rubber Hose.
12. $1\frac{1}{4}\times 8''$ W. I. Pipe.
13. $1\frac{1}{4}''$ Mall. Tee.
14. $1\frac{1}{4}\times 12''$ W. I. Pipe (thread one end only).
15. $1\frac{1}{4}''$ Lock Nuts.
16. $1\frac{3}{4}\times 2\frac{3}{4}''$ Rubber Washers.
17. $1\frac{1}{4}''$ Mall. Tee.
18. $1\frac{1}{4}''$ Pipe Plug.
19. $1\frac{1}{4}\times 4''$ W. I. Pipe (thread one end only).
20. $1\frac{1}{4}\times 4''$ W. I. Pipe.
21. $1\frac{1}{4}''$ Brass Cocks.
23. $1\frac{1}{4}\times 4''$ W. I. Pipe.

COOLING WATER PIPING

4AK. 10. $1\frac{3}{4}\times 12''$ Rubber Hose.
11. $1\frac{3}{4}\times 18''$ Rubber Hose.
12. $1\frac{1}{4}\times 6''$ W. I. Pipe.
13. $1\frac{1}{4}''$ Mall. Tee.
14. $1\frac{1}{4}\times 12''$ W. I. Pipe (thread one end only).
15. $1\frac{1}{4}''$ Lock Nuts.
16. $1\frac{3}{4}\times 2\frac{3}{4}''$ Rubber Washers.
17. $1\frac{1}{4}''$ Mall. Tee.
18. $1\frac{1}{4}''$ Pipe Plug.
19. $1\frac{1}{4}''$ Close Nipple.
20. $1\frac{1}{4}\times 12''$ W. I. Pipe (thread one end only).
21. $1\frac{1}{4}''$ Brass Cocks.
22. $1\frac{1}{4}''\times 4''$ W. I. Pipe (thread one end only).
23. $1\frac{1}{4}\times 3\frac{1}{2}''$ W. I. Pipe.

5AK. 10. $1\frac{3}{4}\times 12''$ Rubber Hose.
11. $1\frac{3}{4}\times 18''$ Rubber Hose.
12. $1\frac{1}{4}\times 10''$ W. I. Pipe.
13. $1\frac{1}{4}''$ Mall. Tee.
14. $1\frac{1}{4}\times 12''$ W. I. Pipe (thread one end only).
15. $1\frac{1}{4}''$ Lock Nuts.
16. $1\frac{3}{4}\times 2\frac{3}{4}''$ Rubber Washers.
17. $1\frac{1}{4}''$ Mall. Tee.
18. $1\frac{1}{4}''$ Pipe Plug.
19. $1\frac{1}{4}''$ Close Nipple.
20. $1\frac{1}{4}\times 12''$ W. I. Pipe (thread one end only).
21. $1\frac{1}{4}''$ Brass Cocks.
22. $1\frac{1}{4}''\times 4''$ W. I. Pipe (thread one end only).
23. $1\frac{1}{4}\times 3\frac{1}{2}''$ W. I. Pipe.

COOLING WATER PIPING

6AK. 10. $2\frac{1}{2}\times 12''$ Rubber Hose.
11. $2\frac{1}{2}\times 18''$ Rubber Hose.
12. $2\times 8\frac{1}{4}''$ W. I. Pipe.
13. $2''$ Mall. Tee.
14. $2\times 12''$ W. I. Pipe (thread one end only).
15. $2''$ Lock Nuts.
16. $2\frac{3}{8}''$ Rubber Washers.
17. $2''$ Mall. Tee.

6AK. 18. $2''$ Pipe Plug.
19. $2''$ Close Nipple.
20. $2\times 15''$ W. I. Pipe (thread one end only).
21. $2''$ Brass Cock.
22. $2\times 4\frac{1}{2}''$ W. I. Pipe (thread one end only).
23. $2\times 4''$ W. I. Pipe.

Oil Cups. Take the large sight feed oiler, and screw it into a hole on top of the engine at *H*, Fig. 713, and fill this cup with **gas engine** cylinder oil through the little screw cap at the side of the top. Be sure to screw back the cap so that the oil will not splash out. You will notice underneath this oil cup a little opening covered with glass, and when the oil is running you can see the oil drop down. There is a double thumb screw which regulates these drops very easily, as you can see by trying it.

See that the cup is adjusted to drop about 15 drops a minute for the first two or three days your engine runs, then decrease the supply to eight or ten drops a minute, which is about what you should use steadily.

Fill the grease cup on the connecting rod with hard oil. Be sure the cup is kept filled with grease, as it lubricates the pin-bearing of the crank-shaft.

To oil the main bearings you will find oil recesses on top of the main shaft-bearing caps. These are filled with cotton waste soaked in oil, and this waste belongs in there. Do not take it out, except to replace it with fresh waste. Keep this waste thoroughly saturated with good lubricating oil.

Be sure to oil the rocker arm at *L* that operates the valves—it is acted on by the push rod, and should run freely. (Fig. 713.)

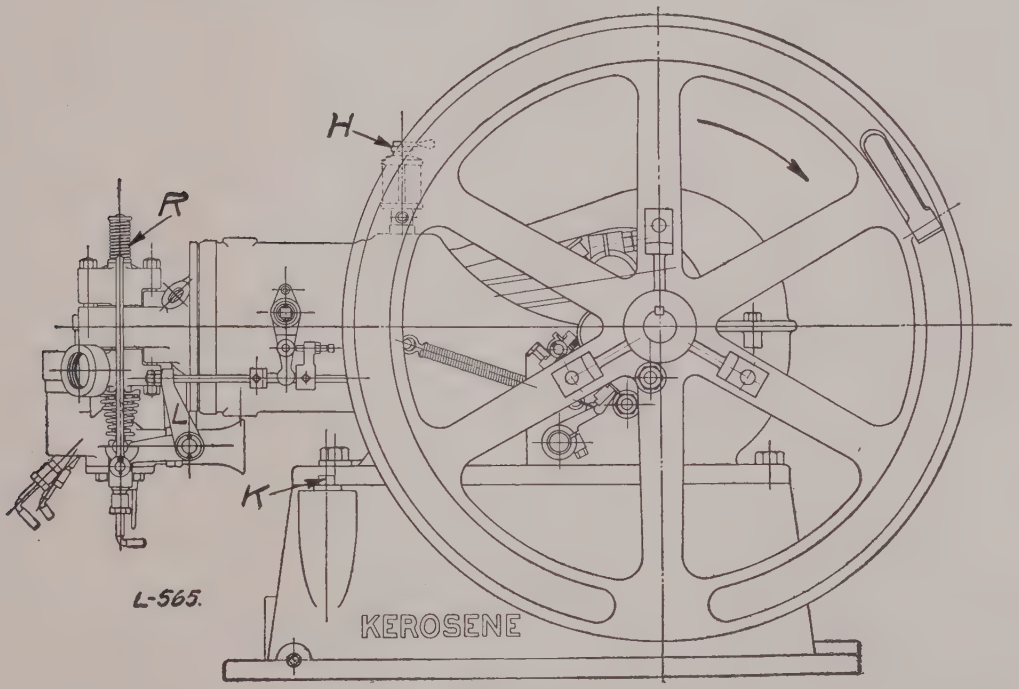


Fig. 713.

To put on the pulley, fit it on the wheel opposite to the wheel which carries the governor. Three bolts are furnished, and are on the pulley, together with the nuts to hold them on the wheel.

Filling with Kerosene. The engine comes all piped up for kerosene. The kerosene is contained in a compartment in the base of the engine. To fill with kerosene, remove the plug at *K*, Fig. 713, place a funnel in the hole, and pour in kerosene, until compartment is filled to within two inches of the top. Put the plug back tightly. Notice that a similar opening is on the opposite side of the engine for water, and the plug for this one marked is *M*, Fig. 715. The kerosene must be put in the opening at *K*. Be sure that no mistake is made and the kerosene put in the wrong side of the engine. **The word "Kerosene," is cast in the side of the sub-base, in which side the kerosene should be put.**

To see whether there is sufficient kerosene in the base put a clean rod down through the hole *K*, Fig. 715. If the rod shows more than 3½ in. of its length wet there is enough kerosene to run the engine; 3½ in. must show, however, before the engine will run. This is because the rod goes down into a small well in the compartment, from which well the kerosene is drawn through the suction pipe. The well is below the level of the bottom of the kerosene compartment and the compartment may be entirely dry while the well shows 3½ in. of kerosene.

The same holds good of the water compartment.

At the opposite side of the engine, in Fig. 715, is the other opening, *M*, mentioned above. This is to be filled with clean water. Both water and kerosene should be strained through a fine mesh of wire gauze before being put into the engine. The water and the kerosene are both used to make the mixture for the engine cylinder, and the reason for straining both is to prevent any dirt from getting in to clog the adjusting valves. The water is fed to the engine with the kerosene to prevent pre-ignition in the cylinder. If the engine knocks or pounds when the connecting rods and bearings are properly adjusted, a slight admission of water will take the pounding entirely away and the engine will run with great smoothness. It will also reduce the fuel consumption of the engine.

The mixer, or carburetor, *N*, Fig. 715, is shown in section in Fig. 716; that is, it is cut through and the drawing is shown as looking into one-half of it. In the section shown in Fig. 716, two reservoirs, *A* and *B*, are shown as being on opposite sides of the central tube *C*. In the actual mixer, they are both on the same side of the central tube, and are only shown opposite in the drawing for greater clearness. The use of these reservoirs is to contain a supply of kerosene and water at the level *DE* shown. Every time there is a suction stroke the engine draws in a charge of air through the opening *F*, which is an open passage from the air into the central tube of the mixer, a slight vacuum being formed above the narrow neck of the central tube *G*. The operation of drawing the kerosene and water up into the reservoirs, and discharging the excess back into the tank below, is similar in both the kerosene reservoir and in the water reservoir. If we describe the kerosene, you will understand both. The vacuum mentioned above, acting through the port *H*, Fig. 716, draws the kerosene up from the compartment in the base; a small spurt comes up through the pipe *K* every suction stroke. This small spurt amounts to a few drops, and does not fill the reservoir *L*, but as soon as the suction stroke is completed the air rushes in through the port *F* into the central tube, and during the remaining three strokes of the engine atmospheric pressure is maintained in the central tube and the reservoir *L* and the few drops which came into the reservoir *L* during the suction stroke, drop down into tube *M*, past valve *N* into reservoir *B*; and after this has occurred a few times the level of kerosene in the reservoir *B* will rise until it begins to flow out of pipe *O*. Any more kerosene coming down from the tube *M* will simply flow out of pipe *O* back into the base of the engine. In this way a constant level is maintained in compartment *B*, which level is slightly below the top of the nozzle *P*. This nozzle is connected by ports with compartment *B*, and has needle valve *Q* controlling the flow of kerosene from compartment *B* to the nozzle.

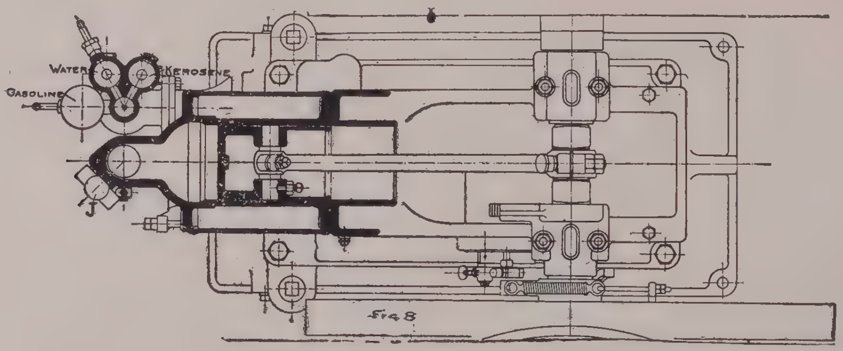


FIG. 714.

As the air rushes past the nozzle in the throat *G*, it sucks this kerosene up through the nozzle with it, making a fine spray. This spray is controlled by the needle valve so that the proportion of air and kerosene are exactly right to cause good combustion in the cylinder. The spray is thus sucked into the cylinder on each suction stroke, and is then compressed, fired, and discharged through the exhaust.

The water comes through the reservoir *A*, through needle valve passage *R*, which is equivalent to another nozzle *P*, except that it enters at the side of the central tube. The water is thus sprayed in with the kerosene, a constant stream being sucked up into the reservoir from the base, the excess overflowing and going back into the base. The check valve *S* allows the kerosene to come up the tube *K*, but not to get back. The check valve *N* allows the kerosene to go down the tube *M*, but not to get back. In other words, the main piston of the engine acts as a pump, valves *N* and *S* being equivalent to its suction and discharge valves.

The cage *T* can be removed from the reservoir by taking out the plug *U*. It can then be lifted out and the valve examined.

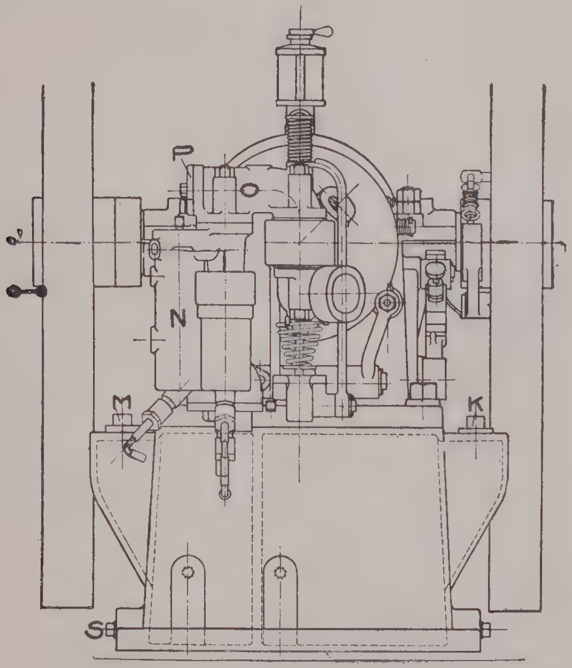


Fig. 715.

The joint *V* is held tight by a set screw *W*. A cone of wire gauze, *X*, is inserted in the upper end of the tube, which helps break up the mixture of kerosene, air, and water into a very fine mist, giving good combustion.

For starting on gasoline, a third reservoir, *Y*, is placed on the side of the mixer, and has a needle valve corresponding to that of the water injection The top of this reservoir is covered by a small cast iron bottle, which can be simply lifted out, filled with gasoline, and inverted into the top of the reservoir. This will maintain the level of gasoline just at the bottom of the bottle when it is in place in the reservoir. It will be seen that this arrangement provides a constant level of water, kerosene, and gasoline in their respective reservoirs; these levels being just below the top of the discharge openings into the central tube, and therefore when the engine stops running there can be no drip of gasoline, water, or kerosene into the mixer. The amount of gasoline, water, and kerosene fed in can be adjusted by their respective needles, ***Z***, ***R***, and *Q*.

The small slide No. 1, on central nozzle *P*, is raised by pushing up on the rod No. 2, when starting the engine. This slide is pushed up into the neck of the mixer at *G* to throttle the flow of air, and cause more vacuum in tube *C* when turning the engine slowly by hand. This draws the fuel in through the gasoline nozzle more readily than when the neck *G* is wide open. In warm weather it is not always necessary to push the slide No. 1 up into place for starting. After starting, it must **always** be pulled down again after the engine has taken an explosion or two.

In the casting *O*, Fig. 715, a second roll of gauze is introduced into the tube with the cap *P*. On taking this cap out the gauze will come with it, as the two are attached. The gauze cone and the roll of gauze above are both used to break up the kerosene into fine mist, and prevent any condensation and drip from the tube.

Starting. Go back and read over the instructions in regard to setting up, so that you know everything is right.

Be sure that there is kerosene and water in the base. Don't take anybody's word for it, but look yourself, and measure it to be sure you know just how much you have. While there is no way for it to leak out, yet if the engine has been damaged in delivery there might be a crack in the sub-base that would allow it all to leak out shortly after you filled it. The engine will not run without fuel.

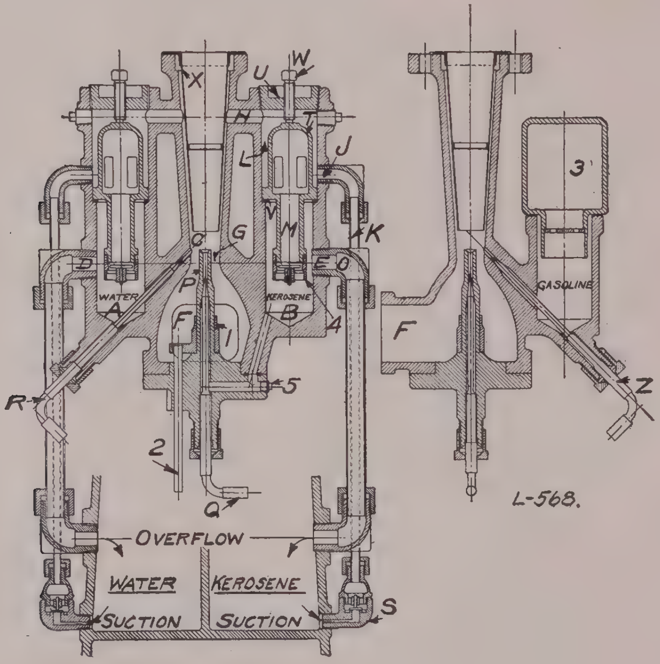


Fig. 716.

Be sure that all the working surfaces are oiled; you cannot get too much oil on a new engine. Be sure that the sight feed oiler at the top is turned on and is dropping. See that the grease cup is turned down so that the hard oil is forced, when warm, through the feed to the bearing.

Oil the small holes on the governor weight and stud. Throw off the switch at *F* on Fig. 707. The cut Fig. 717 shows this switch when it is thrown off, and also how it looks when it is thrown on.

To Start on Gasoline. Revolve the engine to starting position; that is, so the piston is beginning the forward, or explosion, stroke. To do this easily, relieve the compression by holding the inlet valve *R* (top valve *R*, Fig. 713) down during the compression stroke; crank the engine just past the compression stroke to the beginning of the explosion stroke (the switch being open); let the wheel stand in this position for a minute, while you take the gasoline cup, No. 3, Fig.

716, which is part of the mixer, and fill it with gasoline. When it is filled, turn it upside down in the reservoir *Y*, Fig. 716; open the gasoline needle *Z*, Fig. 716, half or three-quarters of a turn. See that the needle valves for water, *R*, and kerosene, *Q*, Fig. 716, are closed. It will be well to remember that every moving part must be oiled by some means. Push up the rod, No. 2, Fig. 716, as far as it will go. Throw the switch on as per Fig. 717; crank the engine over sharply a few times, and it should begin to take explosions, providing the gasoline needle valve *Z* is set right. It may require a little more, or a little less opening of this valve to start the engine, and if it does not go the first time, try a little different setting of this valve. After the starting position of the valve is learned once, it can be put in this position for starting every time.

As soon as the engine begins to explode, pull down the rod No. 2, Fig. 716, at once. If this rod is left up after the engine is started, the mixer will flood; that is, take too much gasoline to make an explosive mixture, and shut down. As soon as the machine is running, the valve *Z*, Fig. 716, can be regulated until the engine is taking the sharpest explosions. When this needle is set right, the machine will run up to speed quickly, and begin to cut out explosions by means of the governor; that is, when it gets up to speed it should take an explosion, cut out several, and then take another. If the exhaust from the engine is black, too much gasoline is being used, and the needle *Z* should be closed a little; if a popping sound is heard at the engine, too little gasoline is being used. A bluish-white smoke coming from the exhaust means too much lubricating oil, but on a new machine this should be left this way for a while, and the oil should be cut down only after the engine has been running a day or so. It is better to have a little too much lubricating oil coming into the engine to start with and only after some considerable running should this be cut down to where it does not show a bluish-white smoke. Eight drops per minute is about the usual feed after the engine is worked in and running right, and with eight drops per minute the colorless exhaust will be found. In starting a new engine, at least fifteen drops per minute should be used.

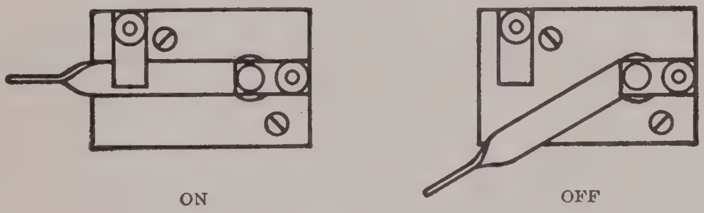


Fig. 717.

The engine is now running on gasoline, and after it has run for a minute or two, the needle valve *Q* (Fig. 716) may be slightly opened. This is the valve which admits kerosene to the engine, and should not be opened until the head of the engine is too warm to hold the hand on. It should be always possible to open up the kerosene valve inside of two minutes after the engine is running. As the valve *Q* is opened the gasoline valve *Z* should be closed. If the engine does not pick up and run promptly on kerosene, it should be run on gasoline a little longer. In opening up the kerosene valve *Q*, it should be opened from about three-quarters to one turn. After the engine picks up and begins to take explosions on kerosene with gasoline valve *Z* closed, the kerosene valve *Q* should then be gradually closed down until the smoke from the exhaust is only the faintest blue.

The engine is now running on kerosene, and the load can be thrown on. When the load is thrown on, after about ten minutes' running, a knocking will be heard in the cylinder, due to the heating up of the engine. This knocking is due to the fact that as the engine heats up, the kerosene in the engine becomes warm enough to pre-ignite because of the compression. At this time the water admission valve *R* should be opened. This valve should be opened gradually, and as it is opened the knocking will become less and less, until the engine is running smoothly. If the needle valve is opened too wide, it will cause the engine to slow down, lose power, and finally stop, and the proper point can be determined by this. In other words, kerosene needle valve *Q* must be regulated to reduce the smoke from the exhaust to the minimum, and still have the engine pulling strongly; and the water valve *R* should be as wide open as is possible, without cutting down the power of the engine. None of the valves, after they are once set right, will require any adjustment for change in load, and the engine will run steadily through all such changes up to its maximum power. The setting of these needle valves which will give these conditions, can readily be found by experiment, and when this is once determined, the valves should be set at this point when running the engine thereafter. The only thing which can change these settings will be dirt allowed to get into the kerosene or water compartments through carelessness. If dirt gets in, the parts should be removed and cleaned. If running water is used instead of a tank, it should be turned on before starting the engine. This water should come away from the jacket warm to the hand. If it is warm enough to be uncomfortable, a little more water should be turned on.

To Start on Kerosene. Read over the instructions for starting on gasoline and be sure you understand them.

On the head of the engine (Fig. 714), is a bulb *J*. This bulb can be heated with a blow-torch to about red heat, and the engine started on kerosene, without batteries, magneto, or gasoline. In this case, of course, the gasoline valve should be closed and the kerosene valve opened three-quarters of a turn from the closed position. Torches are made, and can be furnished, when required, operating with kerosene so that the only fuel needed for starting or running is kerosene. It is much easier, however, to start the machines with gasoline where it is obtainable. To start the engine with gasoline requires only a few teaspoonsful before it will run on kerosene. With a hot bulb, of course, the engine will start off on kerosene. When starting engine on kerosene by means of hot bulb, rich mixture must be used to get the first explosions.

Good results will be obtained by opening kerosene needle valve two turns and raising air throttle.

Hot tube must be heated to a dull red heat.

When using kerosene blast torch the heat of the flame may be increased by raising the air pressure.

To Stop the Engine. Always close water valve R first (Fig. 716); close off the lubricators, Fig. 713; close off the kerosene valve *Q*, Fig. 716. If running water is used for cylinder jacket, close off this jacket water; and lastly, throw off the switch on the battery box *F*, Fig. 707 (the off position is shown in Fig. 717). It is quite necessary to close the water and kerosene valves on the mixer before throwing off the switch so that any water or kerosene remaining in the engine will burn out, and not be left in the engine to rust or cause hard starting.

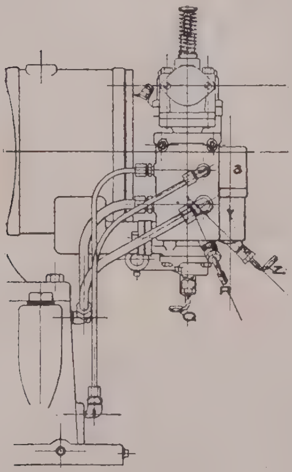


Fig. 718.

What to do if the Engine does not Start. Look everything over carefully. See that you have followed out the instructions exactly. It may be that the gasoline needle valve is too wide open, or too far closed to give proper mixture. If this is not found to be the case, test the spark plug as per instructions on page 624, and if the trouble is not there, test the coil (p. 626).

If the engine, while running, begins to miss fire and explosions occur occasionally in the exhaust pipe, the trouble is usually in the ignition system. If the ignition system is right, and the exhaust becomes too smoky, too much fuel is being used; if a popping occurs in the mixer at the engine, too little fuel is being used. If the spark is weak at the spark plug, but occurs regularly, the trouble may be in the adjustment of the thumb screw on the coil (see p. 623), or there may be a loose connection in the battery box, or the batteries may have run down. If a magneto is used, this is eliminated, for the magneto will always give current, if the brushes are clean. It is possible that the kerosene has been used up, and if the engine shuts down the trouble is usually that it has run out of fuel. This is the first thing to look for when trouble occurs.

If you have plenty of kerosene and water, unfasten the wire at *E*, Fig. 707, and unscrew the spark plug, taking it out of the engine. Then connect the spark plug up again with the wire and lay the plug on its side, on the metal of the engine, being sure that the point does not touch any metal, and that the wire going to the plug does not touch the engine. Throw on the switch again and turn the fly-wheel slowly to the right. When you reach the sparking point (which may take two turns of the wheel), you should hear a buzzing noise and there should be a stream of sparks two or three times the size of a pin head, between the point and outside metal of the spark plug. If the plug sparks down the outside or high up on the inside, or the spark does not jump between the point and the metal of the plug, the plug is dirty or the insulation is cracked. Take your plug apart and clean (instructions, p. 624). If you have the switch on, the connecting wires tight, and have followed the above instructions and on turning the fly-wheel over to the right fail to hear the buzzing noise from the spark coil, there is a loose wire somewhere in the cells, the batteries are exhausted, or the spark coil needs adjustment. The fault lies so very seldom with the spark coil that we do not recommend it to be touched at all, except as a last resort, as in nine cases out of ten the fault lies elsewhere. Do not touch your coil unless you positively know that the trouble is there and nowhere else. The chances are that it is somewhere else. If the trouble was with the spark plug, and you now get a good spark, put it back, connect the wires, and your engine will probably start right off.

If you get no spark, and the plug appears to be right, look at the batteries. Make sure first that there is no loose connection, that all thumb nuts are tight; a loose wire anywhere will be enough to stop the engine. If you have recently changed cells, look over the wiring and be sure that it is wired up right, as per Fig. 707. If everything is right, take your battery tester, and test each cell for its amperage—each dry cell should show at least 1½ volts and 15 amperes. It will not do satisfactory work unless it shows at least 8 amperes. If it does not, throw it away and put in a new cell. You are simply destroying your good cells by leaving this bad one connected up with them. Every user of a gasoline engine should have a battery tester.

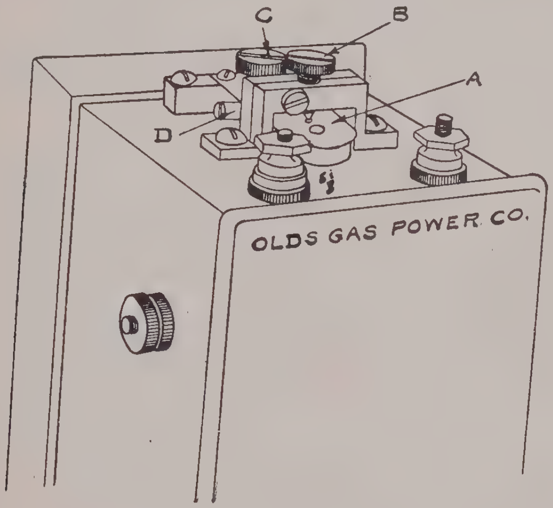


Fig. 719.

If the battery is all right, if the wires are right, and the plug is all right, squirt a little gasoline into the spark plug, put it back, connect the wires, and turn your engine over two or three times. It may be that there has been a little clog in the gasoline pipe, and if the engine will take one or two shots from priming in this way it will often clear itself. If it does not, look at the mixer, it may not be getting gasoline. The feed may be stopped up, though this is unlikely, as the engine has undergone three separate tests by us in our factory before it was sent out. Try the engine with the needle valve less widely open; it may be that it is flooding because of some small dirt obstruction—if it does not start then, let it stand a few minutes, try with the needle valve further open than ordinary—if it still does not start, look at the coil, and turn the fly-wheel very slowly to the right; the coil is ordinarily the last thing to get out of order, and for this reason should never be touched until it is positively known that the trouble lies here. If the spark coil does not give out a buzzing noise when the sparking point is reached, loosen the set screw on the side of the frame just below large screw *B*, Fig. 719, turn screw *B* down a little and try again. If it then does not start, turn down screw *B* a little further, then try it, always remembering that the vibrator *A* must still have space in which to move. If it refuses to work, your batteries are dead, or you have a loose wire, or your spark coil has been spoiled by having a nail driven into it, or being injured in transit or unpacking. The chances are ninety-nine out of one hundred that it is a loose wire or a dead cell. Be sure that you have tested the batteries and wires before laying the difficulty to the coil.

If magneto is used the spark can be tested by taking the plug out, connecting it to the wire again, and laying it on some metal part of the engine. Crank the engine over **sharply,** watching for the spark between the points of the plug. If the spark does not occur, the same reasons may hold good as in the case of the batteries, except that in place of the batteries being run down, the brushes of the magneto may be dirty.

How to Clean a Spark Plug. If the spark plug is dirty, wipe it off carefully with a dry cloth, and scratch the points with a knife so that they are bright and clean. Remove all carbon and oil from the porcelain. The points should be about $\frac{1}{32}$ of an inch apart; if they are not, bend them carefully with a pair of pliers.

If the spark takes place down the outside, or anywhere except directly between the point and the side, it is dirty or the insulation is cracked. Take the plug apart. You can do this easily by using two wrenches. Be sure that the insulation is not broken. If it is, you must get another plug. Either put in new porcelain or get a new plug.

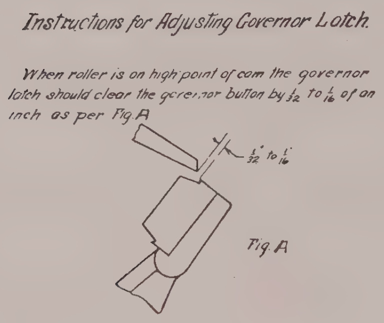

Instructions for Adjusting Governor Latch.
When roller is on high point of cam the governor latch should clear the governor button by $\frac{1}{32}$ to $\frac{1}{16}$ of an inch as per Fig. A
$\frac{1}{32}$" to $\frac{1}{16}$"
Fig. A

FIG. 720.

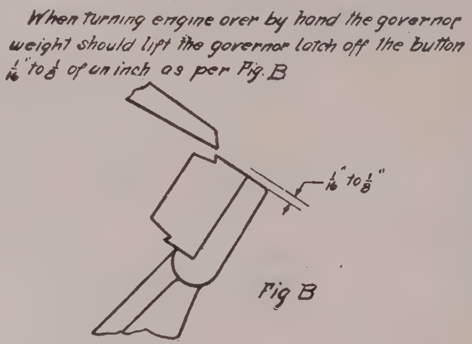

When turning engine over by hand the governor weight should lift the governor latch off the button $\frac{1}{16}$" to $\frac{1}{8}$ of an inch as per Fig. B
$\frac{1}{16}$" to $\frac{1}{8}$"
Fig B

FIG. 721.

Instructions for Cleaning Mixer and Its Valves. In Fig. 716, to clean the mixer, unscrew the screw *W*; take out the plug *U*; lift the cage *T* out of the reservoir; and unscrew the cap No. 4 from the end of the cage *T*. The small valve *N* can be easily cleaned. Put the valve and cap No. 4 back into place; and the cage *T* back into the reservoir, after having cleaned out any sediment that may be deposited in the bottom of the reservoir. Be sure that the little gasket *V* that the cage rests on is clean, as this joint must be tight after the plug *U* and set screw *W* are in place and screwed down. The valve *S* can be cleaned by taking off the back nut which holds the pipe tight. Unscrew the whole valve from the base, and unscrew the cover of the valve from the body of same. The valve can then be thoroughly cleaned on the inside, put together and screwed into place again. To clean the needle valves from the reservoirs to the mixer, they can be simply unscrewed until they come out. It is well to wash out the holes with gasoline or kerosene when the needles are out so that any dirt in the holes will not be left in the threads. The small plug No. 5, Fig. 716, can be removed and the hole to the needle valve thoroughly cleaned. When this has all been done the cone *X* should be removed and heated in a flame to burn off any lint which may have accumulated on the gauze. This gauze should be heated in a clean flame so as not to leave a carbon deposit. The cover *P*, Fig. 715, can be removed. To this cover is attached a roll of gauze, and this should also be heated slightly, enough to burn off the lint, but not to melt out the babbitt in the cap, afterwards being put back into place. Where an engine has worked in a dusty place, such as running a threshing machine, a very considerable amount of dust is carried in the air, and clogging may occur once in a while. It will usually show by loss of power in the machine; when all the other parts of the engine are in good working order.

How to Adjust the Governor Latch. If the engine does not govern properly, look at the governor latch. When the roller is on the high point of the cam, the governor latch should clear the governor button by $\frac{1}{32}$ to $\frac{1}{16}$ of an inch. (Fig. 720.) When turning the engine over by hand, the governor weight should lift the governor latch off the button $\frac{1}{16}$ to $\frac{1}{8}$ of an inch. (Fig. 721.)

How to Adjust the Governor to Regulate Speed. In order to regulate the speed of the engine it is necessary to adjust the governor. (Fig. 722.) Screw up the nut M on the governor weight to make the engine run faster, and unscrew it to make it run slower. If you wish to change the speed of the engine slightly, increasing or decreasing it 50 to 75 r.p.m., this can be done while the engine is running by screwing the thumb screw N (Fig. 722) out to increase, and in to decrease the speed. Only a very slight turn is necessary.

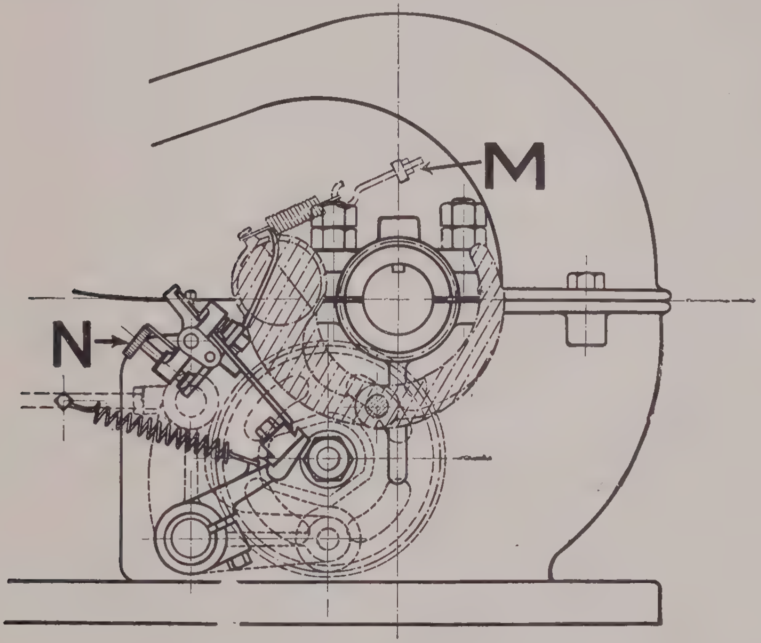


Fig. 722.

How to Tighten the Connecting Rod. If there is a pound or knock in the connecting rod it should be tightened either at the piston end or at the crank, or both. To do this you will have to remove the oil shield between the wheels by unscrewing two little studs on either side, which will allow you to take the shield off.

In the tool box you will find a special length socket wrench (No. 18) which fits the adjusting nut on the top side of the connecting rod at the piston pin inside the piston. Place this socket wrench on the nut, take a wrench and turn it to the right, turning the socket wrench to the right, being careful not to get it too tight. There must be a little play in this connecting rod, but not too much. You can tell how much there is by taking hold of the connecting rod and shaking it. It must be a trifle loose. Screw this up enough to take up the play which makes the noise. It must not knock when running.

On the other end of the connecting rod you will find a split box or bearing which is adjusted by four nuts, two top and two bottom. Loosen the lock nuts there and screw up the others. If you cannot get this tight enough, take off the cap and cut down the leather shims found between the cap and the connecting-rod, then screw down nuts until tightness can be felt on turning the crank, then let back a little to loosen. See that the bearing does not heat; if it does the box is too tight and should be loosened further. Be sure to keep these two adjustments tight enough so that you will hear no knocking.

Compression. In turning over the wheel when the spark plug is screwed in, you will find one point on every second turn where the engine can only be turned with difficulty. This is caused by the compression which you should always have. If you do not have it, the rings on the piston are gummed up and stuck. Give your engine plenty of oil, and the rings will probably free themselves, and your compression come back to what it should be. This will often be found to be the case where an engine has been standing for some length of time.

The rings become gummed or rusted into the piston and do not spring out as they should and close the space between the cylinder and piston walls. If the compression continues bad, disconnect the connecting rod and pull the piston out of the engine, being careful not to injure the piston or break the rings. If you have been using good oil your piston will be bright and smooth—bad oil, dark and gummy, with the rings stuck up.

Clean the piston thoroughly with kerosene, be sure that the rings are free before putting back, and oil thoroughly with good gas engine oil. In putting the piston back into the cylinder, be careful not to pinch or break the rings. If the rings on the piston are loose and the piston is bright and smooth and evidently in good shape and you do not get compression, it is because the valves are not closing properly. It may be that some dirt has gotten underneath them, or a little piece of waste, which prevents their closing.

It is best not to tamper with the setting of the valves as they cannot get out of order under ordinary circumstances. The accompanying figure (Fig. 723), will show you how they should be set, if you ever have occasion to take the engine down.

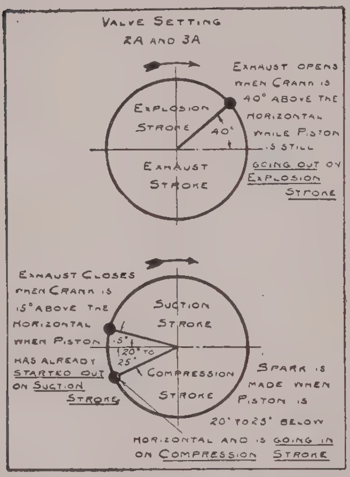

VALVE SETTING
2A AND 3A
EXPLOSION STROKE
40°
EXHAUST STROKE
EXHAUST OPENS WHEN CRANK IS 40° ABOVE THE HORIZONTAL WHILE PISTON IS STILL GOING OUT ON EXPLOSION STROKE
EXHAUST CLOSES WHEN CRANK IS 15° ABOVE THE HORIZONTAL WHEN PISTON HAS ALREADY STARTED OUT ON SUCTION STROKE
SUCTION STROKE
15°
20° TO 25°
COMPRESSION STROKE
SPARK IS MADE WHEN PISTON IS 20° TO 25° BELOW HORIZONTAL AND IS GOING IN ON COMPRESSION STROKE

FIG. 723

How to Adjust Spark Coil. If your batteries have grown weak and the buzzer fails to work, you can often get a spark and go on running for a short time until you can get new batteries, by loosening the set screw on the side of the frame below set screw *B*, Fig. 719, and then turning down screw *B* and decreasing the space between vibrator *A* and screw *B*. Always remember that the longer the space between the vibrator *A* and the point of screw *B*, the stronger and hotter will be the spark in the engine, and the better the engine efficiency and power. Also, always remember that vibrator *A* must have room to move, and that if *B* is screwed down tightly the engine cannot run at all, and that the longer the "throw" which you can give the vibrator *A* the better the engine will run.

On putting in new batteries unscrew screw *B* on the coil until the current will just throw the vibrator. You can find this point by turning the engine over slowly, everything being connected up and the switch on.

After months of running the point on screw *B* may become battered—take it out and smooth the point up carefully with a file, so that it gives a good even contact with a clean flat point about $\frac{1}{32}$ of an inch in diameter. It is well to keep the top of your coil covered up, as the dust may get in and fuse on to the point *B* and stop your engine.

General Suggestions. Keep Your Engine Clean. Before starting the engine always oil the piston and parts with a hand oil can; see that plenty of standard gas engine oil is in the oil cups—do not use any other kind of oil but gas engine oil.

Always see that all the nuts and screws are kept in their places and are tight. A loose nut spells trouble.

Be sure to oil the rocker arm and valve occasionally. (*L*, Fig. 713.) If this arm does not work freely at first, wash it with a little kerosene, as it may get gummed; then oil it, being careful that all kerosene is out.

Never touch the spark coil until you are sure the trouble is there and have been over everything else. It is the last thing to get out of order, and the last thing that should be tampered with.

If the engine turns hard, it is not getting enough lubricating oil, or the oil is poor.

Put in no screws or nails that will reach through the outer casing of the battery box and the spark coil box; they will ruin the coil.

Keep the coil in as dry a place as possible; moisture will cause leakage of current.

Never leave the switch on. If the switch is left on, it will use up a good set of batteries in a few hours.

In making adjustments never leave the engine in such a position that the buzzer will keep working; it will use up your batteries very quickly. In testing or looking for any difficulty, if you wish to try the buzzer, do not keep it buzzing any longer than absolutely necessary.

Keep an extra spark plug on hand, clean, perfect, and ready for use—it will save you time and trouble.

In cleaning up a spark plug, be sure that the insulation is not cracked.

If a battery becomes exhausted, it will start the engine all right in the morning, but will run down before the engine has run long.

Test your batteries. After the engine has stood for a time the batteries get stronger, but they will die on very short running.

If the engine makes a popping sound at the point of air intake, or air throttle, it is not getting enough kerosene. Open the needle valve slowly until this stops.

If an explosion takes place in the muffler, do not be alarmed; it only means that there is too much kerosene, or the spark is poor.

If the exhaust is bluish-white, do not give the cylinder quite so much lubricating oil or kerosene; it may be caused by either. Always remember, however, that of the two faults it is better to give an engine too much lubricating oil than too little, and especially remember that a new engine needs plenty of oil until all the parts are worn perfectly smooth.

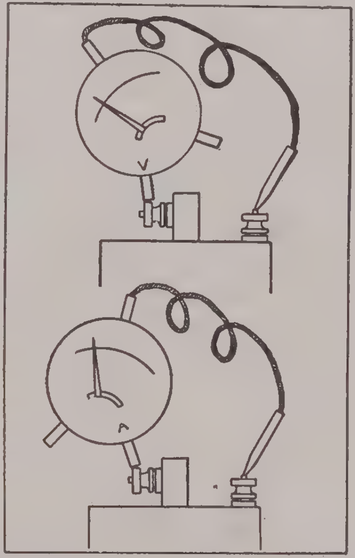


FIG. 724.

Every user should have a battery tester. This will enable you to test your cells yourself, and you will always know whether you have a good cell or a poor one. The method of applying it is to place one point on one of the screws that comes out of the cell, and the other on one of the brass tips, press the button on the tester, and note the registration of the amperes. If it registers over 10 or 12, the cell is all right. Test all your cells in the same way. It is not necessary to disconnect them from each other to do so. Remember that if even one cell is weak it reduces the efficiency of the whole set. Throw it out and get another. If you cannot get another immediately, throw it out and run with the remaining cells.

A battery will wear out in time whether it is used or not, so it will not do any harm to have some fresh ones on hand. With this battery tester you can also test the cells that you buy, and be sure that you get good live ones. Do not buy dead ones, as they are worthless to you.

Keep your engine clean. You cannot get good work from a dirty engine.

Use only good gas engine oil. Steam cylinder oil, or sewing machine oil, or any oil that is not made for heat, **will not do.**

Always see that all screws and nuts are tight before starting up for a day's run. They will work loose on any machine, and keeping them tight will greatly prolong the life of an engine.

In Cold Weather. Always drain the engine when through running, and avoid all possibility of breakage because of freezing. The engine may be harder to start in cold weather than in warm. In bitter cold weather the metal in the mixing chamber gets so cold that the gasoline does not vaporize, but remains liquid and the mixture is not right for the first explosion. If the engine does not start, take out the plug and squirt a little gasoline into the plug itself. The type "AK" engine will always start readily in any kind of weather if it is in adjustment. In bitter cold weather when gasoline will not vaporize, the above suggestions may be of value.

If it is near your house, an equally easy way is to pour some warm water over the inlet chamber where the fuel goes into the engine—it will then always start readily.

Remember that in cold weather lubricating oil does not run readily, and take particular care to see that the sight feed oiler at the top of the engine is letting in the usual quantity of oil.

The oil cup should be kept **wider open** during cold weather to insure proper lubrication. Bluish-white smoke from the exhaust will tell you when you have too much. Failure to give oil enough may cut your piston badly in a few minutes.

In order to drain the water out of the engine, open plug at (18), Fig. 712. The water jacket of an Olds engine is removable and can easily be replaced if broken by freezing, but to let it freeze is simply carelessness. If you drain your engine it cannot happen. Water must also be drained out of the base, at point *S*, Fig. 715.

Portable Engines. These instructions apply to portable engines, but in addition there is a grease cup on the rotary pump which must be kept full of grease and in working order. This should be taken care of in the same manner as the grease cup on the engine, saving you the bother of greasing in the old-fashioned way.

In General. Do not be afraid to write to us in case of any difficulty, but above all study this instruction book carefully until you have mastered it and understand the engine completely.

The Olds is the simplest form of engine on the market, and the easiest kind of engine to take care of.

One of our best repair men goes out on the road with the sole equipment of a lead pencil, and we have never yet found him unable to make good. The only thing that can put an engine out of business if you fully understand these instructions and follow them—is the actual breaking of a part.

If you are stuck, ask the nearest agent, who can probably set you straight. If he cannot, write us; that is what we are here for—to see that Olds engines run and run right, and we are always glad to know how you are getting along and see that your engine is giving good service.

Above all, remember that the engine has run, and will run, and it is simply a matter of getting everything right. If everything is right, the engine must go. The repair man with the lead pencil is an object lesson; he knows where the trouble is, and it is only a question of slight adjustment. Remember that the Olds engine is the simplest gas engine made; it takes only a little patience and common sense to work out your difficulties yourself. There are thousands of them being used every day in a dozen different countries without any trouble, and there is no reason why yours should not work as well as the rest.

Now, before trying to start your engine, read this book through again.

OLDS GAS POWER COMPANY,
Lansing, Mich., U. S. A.

2. Directions for Starting the Hornsby-Akroyd Oil Engines

De La Vergne Machine Company

1. On engines up to 25 H.P., close the valve supplying water to the vaporizer entirely; on larger engines, leave the valve slightly open.
2. Start the lamp for heating the vaporizer. Place it underneath the vaporizer, allowing it to remain for a sufficiently long time to heat the vaporizer.
3. Fill oil cups, and oil the engine thoroughly.
4. Set the oil pump to give a slightly longer stroke than No. 1 gauge (or full load pump stroke).
5. Open the overflow valve at the governor; work the oil pump by hand until the oil is seen through the glass gauge to pass freely from the overflow valve. Then close overflow valve.
6. Pump a little oil into the vaporizer; if the vaporizer is hot enough, on opening the test cock on top of cylinder, oil vapor will issue in which case cock must again be closed. *It is important that the vaporizer be hot enough, before starting the engine.*
7. Give one or two strokes to the oil pump by hand, and turn the fly-wheel over until ignition takes place.
8. *Reduce oil pump to No. 3 gauge until load is put on.*
9. Open the valve on the water supply to the vaporizer slowly and carefully, so as to prevent rapid contraction of the vaporizer walls.
10. Take heating lamps off.

Directions for Cleaning the Vaporizer of the Hornsby-Akroyd Oil Engine. If the engine runs on kerosene, carbonization in the vaporizer takes place only if the engine is overloaded or not handled properly. If the engine runs on fuel oil or crude oil, the vaporizer must be cleaned after a running period of between 24 hours and two weeks, depending entirely upon the quality of oil used, and conditions under which the engine is run.

When the vaporizer cap is removed for cleaning, no open flame should be brought into, or near to, the open cylinder, and the crank should be in such a position that the exhaust valve is open. This precaution will allow any explosive mixture which might remain in the cylinder to escape through the exhaust pipe.

3. Operating Westinghouse Vertical Gas Engines—Instructions to Engineer

Care of Engine. 1. Maintain crank case oil level at the parting of the rod-bolt nuts on all engines up to and including 11×12″ three-cylinder, $\frac{1}{2}$ in. above the parting of the nuts on 13×14″ three-cylinder, $\frac{1}{2}$ in. up to the lower brass on 15×23″ and 18×22″ three-cylinder engines, and about 3 in. up the lower brasses on 25×30″ three-cylinder engines.

2. Use Atlantic Refining Company's Westinghouse gas engine oil, or oil of same quality, in two main oil cups and in crank case.

3. Use Standard Oil Company's Renown engine oil, Leonard & Ellis Red Star machine oil, or other oil that is free from a gumming tendency, on other parts of engine. If mixing valve or ignitor stems have a tendency so gum up, use a half-and-half mixture of this oil and kerosene (coal oil) to keep them free.

4. Governor and ignitor stems *must work perfectly free.*

5. Igniter trip stem nuts should be so set that, when the stem has just dropped off the cam, the buffer washer holds trip stem $\frac{1}{16}$ in. off cam, and when cam has pushed trip stem farthest out, there is $\frac{1}{32}$ in. to $\frac{1}{16}$ in. clearance between igniter arms and trip stem nuts.

6. Never run engine without cotter pins in inlet valve stem and igniter stems.

7. Keep exhaust valves tight. When allowed to leak they become burned. If valves are tight the compression will be very noticeable when it is attempted to turn engine forward rapidly by hand with exhaust starting levers in vertical position. After grinding exhaust valves, always make sure that there is from $\frac{1}{64}$ in. to $\frac{1}{16}$ in. clearance below the lower end of upper exhaust valve stem, according to size of engine, when on idle part of cam. When this is adjusted correctly, the valves will seat quietly but firmly.

8. Keep igniters clean and dry, and do not allow them to leak, or they will heat and be damaged.

9. Remove and examine condition of igniters at least once a week and of exhaust valves at least once a month. This cannot receive too much attention. When any leak appears the parts must be immediately ground in to prevent ruining them. A good engineer will examine these parts often enough to have them always in good condition, as it is easily done.

To Start Engine. 1. Make sure of gas at engine by lighting it at pet cock close to throttle. It should not burn violently, for that indicates that regulator gives too much pressure. It should burn about like an ordinary gas jet.

2. Set mixing valve adjustments in best position for starting. This should be carefully determined after installation, and marked. The best running mixture should also be determined and marked.

3. Exhaust starting levers must be in slanting position.

4. Start oil cups.

5. Open gas valve and immediately turn engine forward briskly a number of turns, by hand, in the case of small engines, or by dropping the air starting valve stem onto its cam and then opening the air cock, in the case of larger engines. As soon as explosions are obtained and speed is sufficient, shut off air, throw exhaust starting levers to vertical position, thus giving full compression, and the more powerful explosions will quickly speed the engine up. If these levers are thrown too soon, the compression will stop the engine.

6. As soon as engine is started, turn on jacket water. This must not be delayed, but it should not be done before starting, except in cases where jackets have been drained, as a warm engine starts much easier than a cold one.

7. Move mixing valve adjustment levers out together to running position and put on load.

8. Engineer must satisfy himself that everything is normal about engine before going abcut any other work.

Running Engine. 1. Main bearings should not get hotter than crank case.

2. Engine should receive such attention as a steam engine of same size requires. In the case of small engines, a good engineer, by using his intelligence and keeping engines in good condition, will have much time to spare for other work, provided the same can be done as a side issue.

3. Jacket water should leave engine not hotter than 150° F., nor colder than 98° (blood heat), to obtain the best results.

4. Engine should not misfire. Open pet cocks above exhaust valves to determine this point.

To Shut Down Engine. 1. Shut off jacket water.

2. Shut off gas.

3. Shut off spark. (Be sure that battery is never left short-circuited.)

4. Throw out starting levers.

5. Stop two-cylinder engines with both exhaust valves closed; three-cylinder engines with two valves closed.

6. Shut off oil.

7. Keep engines always warm, if possible, but if there is danger of freezing, drain water from jackets and pipes.

4. Instructions for the Operation of Pressure Gas Producers Manufactured by the Gasmotoren-Fabrik Deutz[1]

Operating the Steam Boiler. The water level in the boiler must be kept at the prescribed height, and the pressure maintained at about 4 atm. (60 lbs.). Loading or otherwise interfering with the valves is absolutely prohibited. The best fuel to use is gas coke, which burns with little tendency to soot or smoke and makes a fire easy to control. The use of other fuels is, however, of course quite permissible.

The exterior of the boiler and especially the steam pipe from the boiler to the injector at the producer should be carefully covered with some non-conducting material.

Cleaning openings (hand holes) through which any mud or scale may be removed are found both in the top of the boiler and near the bottom of the (vertical) shell.

All of the apparatus feeding the boiler should be tested at least once a day to see that they operate properly. The feed water should be as soft as possible, not showing much tendency to throw down scale. Depending upon the quality of the water, the boiler should be cleaned and washed out every one to four months.

Starting the Producer. To start the producer from the cold, open the valve in the purge pipe, the poke-holes in the top, the inside and outside covers of the filling hopper and the ash pit doors. (In case of coke producers also the protecting and fire-doors).

Through one of the fire-doors start a wood fire and throw in from time to time small charges of coke or anthracite, but only after the preceding charge has burned through to a bright glow. After the glowing fuel column has reached a height of about 6 in., close all of the doors, after having luted the edges of them with a little soft loam or clay to make them air tight. Before closing the fire doors of the coke producers do not fail to first put the inner protecting door in place.

Next close the filling hopper and the poke holes and then turn on the steam, the pressure in the boiler having been raised in the mean time. The air blast serves to produce a stronger blast and causes a more rapid burning up of the fuel column. More fuel is added from time to time, as the glow appears at the top of the charge in the producer, until the column has reached the bottom of the fuel magazine, which may be ascertained by introducing a rod through one of the poke holes. This is the height at which the level of the fuel should be maintained during operation. After the column has finally burned through, an attempt should be made to light the gas at the test opening above the purge valve. If the gas burns with a steady flame, without lifting away from the opening or going out, for about 5 minutes, it is ready for use. Before turning the gas into the cleansing apparatus and the gas holder, make sure that all seals (at the scrubber, sawdust purifier, etc.) are properly filled with water, and that the water supply is turned on for all parts requiring it (top of producer, scrubber, gas pipe leading to seal box, etc.). The supply should be so regulated during operation that the water leaves the scrubber as cool as possible, the seal box lukewarm and the top of the producer hot. All valves to the gas holder may now be opened, next close the valve in the purge pipe. In order to test the quality of the gas during operation a quarter inch burner should be connected to any suitable place in the gas pipe and a flame should be kept constantly burning.

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[1] See illustration, Fig. 407.

Stopping the Producer. If the gas make is to be interrupted for a short time only, open first the valve in the purge pipe, next shut off the steam. During such brief periods the fire in the boiler should only be banked, so that operation can be resumed at a moment's notice.

If the producer is to be banked for a longer period, say over night, shut off the steam, close the valve between seal box and scrubber, and open the purge valve. In this way there will be a constant small air draft on the producer, sufficient to keep glowing the constants of the producer. Fire and ash doors remain closed. The fire under the boiler should be dumped. At starting rebuild this fire, and if necessary also clean the grate of the producer.

Cleaning the Producer Grate. The best time for cleaning the grate is in the morning just before starting. Fuel which clinkers badly requires cleaning oftener.

The grate can be cleaned only when the gas-making is interrupted. The steam must therefore be shut off, the purge valve must be open, while the valve between seal box and scrubber is closed. Open the ash door, carefully clean the grate of ash and clinker by means of poker and hook, and take care that as little as possible of the glowing fuel is drawn down and out with the ash.

In the case of coke producers a false grate is used to help in the cleaning of the grate. For this purpose, after the fire doors and the inner protecting doors are removed, a couple of transverse bars are laid in recesses in the fire-door frames and across the doors. Bars are then forced through the fuel column, resting at each end on the transverse bars. The grate may now be cleaned without the danger of having the entire column come down, since the latter now rests on the false grate. After cleaning, the bars are again drawn out, the transverse bars lifted away, and the protecting and fire-doors are replaced.

In the case of fuel showing a tendency to clinker, it is well to poke down any clinker adhering to the sides of the producer, through the openings in the top, before again starting up.

Dumping the Fire out of the Producer. After an operation lasting several weeks, and especially after the quality of the gas has shown a steady falling off for some time, the producer should be put out of commission and thoroughly cleaned.

After the fire has been drawn, any ash or clinker adhering to the lining must be removed by poking through the top openings, taking care, however, to injure the brickwork as little as possible.

If the lining is so far worn out that gas can find its way through it and between the sand filling and the exterior shell, the only remedy is to entirely replace the lining.

Regulating the Rate of Gasification. To regulate the amount of gas made, a shut-off valve in the steam line leading to the injector is connected by a light chain with the bell of the gas holder. As long as the gas holder is not completely filled, the valve is wide open, but when the bell is near the top, its further rise closes the valve. Gasification then stops until the drop of the bell permits a weight to open the valve again. The rate of the gasification may within narrow limits also be controlled by adjusting the valve in the line from the steam boiler.

Charging the Producer. To charge the generator, fill the hopper and close the outer cover. Then, by raising the counterweighted lever, open the inner cover, and the fuel will drop into the producer. After this raise the outer cover again and ignite the gases caught in the hopper between the two covers. Do not replace the outer cover, but let any gas that escapes around the edges of the inner cover burn in the hopper, in order to prevent the noxious gases escaping into the room.

Seal or Hydraulic Box. The seal box serves to catch and deposit any coal dust which comes over with the gas in quantities, depending upon the nature of the fuel. The deposited slush must be removed about once in 14 days. *During this operation the purge valve must be open and the valve between seal box and scrubber must be closed.* Taking the cover off the seal box gives access to the interior. After cleaning, replace the cover and fill the box with water until it appears through the overflow.

Sawdust Purifier. (These are furnished only in plants using bituminous coal.) Each of the grids should be covered with sawdust to a depth of about 4 in. The charges must be renewed (every 8 to 14 days) only when the valves ahead of and beyond the purifier are closed. Open next the ventilating valve on the purifier and then take off the cover. After renewing the sawdust, open first the valves in the gas line, and leave the ventilating valve open until the gas has displaced all of the air in the purifier.

Scrubber. The scrubber is filled with pieces of coke large enough to prevent their dropping through the grate at the bottom. The water supply at the top of the scrubber should be regulated so that there will be a stream at least the size of a lead pencil flowing from the seal box.

The scrubber should be cleaned at intervals of from 9 to 12 months, depending upon the length of daily operation. For this purpose the plant must be shut down. Close the valve between scrubber and seal box, shut off the water supply and take off both the top and bottom cover plates of the scrubber. The latter should then be allowed to stand for several hours in order to let the gas escape. Next draw out the coke through the bottom opening. After the new filling has been put in, replace the covers, renewing the packing wherever it is defective. The coke of the old charge may be used under the boiler.

All cleaning operations should be undertaken only in the day time, and care should be taken to see that there shall be no fire, open flame, light, or smoking in the room during that time. The cleaning should always be done by at least two men, and on account of the dangerous, poisonous quality of the gas, the greatest care should be exercised, especially to see that there is ample opportunity to get fresh air.

Starting up the Plant after Installation or after Cleaning. *If the scrubber, gas holder, or piping has not yet been in operation, or if they have been open to the air for cleaning purposes, the spaces will be filled with air which with the first gas made will form an explosive mixture.* If therefore a plant is to be newly started, it becomes necessary to completely replace the air and after it the air-gas mixture by pure gas. To this end open first the valve in the purge pipe, so that the gas first made may escape to the open air. Next close this valve and fill the gas holder, then stop the steam blower, and allow the gas holder to discharge through the valve at the extreme end of the gas main. Repeat this operation two or three times, depending upon the size of the holder and the length of the main. *Take care that there shall be no open flame of any kind near the discharge opening. After this process, the gas may be regularly used for the purposes desired.*

Gas Holder. The counterweight for the gas bell should be of such weight that, when the bell sinks, the gas in the mains between the holder and the place of consumption is under a pressure of about $1\frac{1}{4}$ in. of water. Since the water in the holder slowly evaporates, provision must be made to supply the deficiency from time to time. Once a year the entire filling of water should be renewed.

When the water is being completely drained off, provision must be made to supply air through a valve either in the gas main or in the top of the bell. The reason for this is that the drawing away of the water may produce so high a vacuum in the bell as to cause the atmospheric pressure to crush it in.

The water collecting in the traps of the gas main should be pumped out once a week. If the gas holder is not protected against cold, and if there is danger of freezing, a "convection" heating furnace must be installed. In order to obtain a good circulation of the water, the level in the holder should stand about 4 in. above the opening of the pipe supplying the warm water. This point is important and should be especially looked after. Where no very severe freezing is to be expected, it is also possible to protect the holder by means of some covering and to supply steam from the boiler to keep the temperature of the water sufficiently high.

Special Notice. Producer gas, on account of its content of carbon monoxide, is very poisonous and seriously attacks the organs of respiration. All valves and cocks in the gas mains, as well as the filling hopper covers and poke hole stoppers, should therefore be kept as clean and gas tight as possible. They should all be tested at least once a month and all leaks discovered must be remedied. Since the odor of the gas is nowhere near as strong as that of illuminating gas, any leaks occurring are not easily noticeable, and the examination of the entire plant for leaks should therefore be carried out with special care.

5. Instructions for the Operation of Pressure Gas Producers made by Körting Bros., Hannover [1]

1. **Steam Boiler.** The steam boiler must furnish to the injector a constant supply of dry steam. The steam pipes leading to the latter should therefore be well covered. The best fuel to use is coke. If other fuels are used care should be had to see that the superheater coil placed in the upper part of the producer does not come in contact with the fire, as it may be seriously injured thereby.

The smaller sizes of boiler are fed by hand pump from a feed-water tank, the larger sizes are furnished with two Körting Universal injectors. Special directions are furnished for the handling of the latter. The boiler should be put into operation before starting the fire in the producer.

2. **Gas Producer.** To start the producer, open the ash pit doors and the valve in the purge pipe, and build a wood fire on the grate. As soon as possible add coal, which is best done through the filling hopper, and slowly build up the fuel column as high as the natural draft available permits. When the column has reached a height of about 6 to 8 in., close the lower doors, start the steam blower and add fuel as rapidly as possible without smothering the fire.

The lower doors should be made gas tight with loam or clay. The use of a little coke will make the starting of the fire easier. The coke or coal used in regular operation should be dry, because water lowers the quality of the gas made.

When the height of the fuel bed has reached say 25 in., no more fuel is added until the top layer commences to glow. The interior of the generator may be watched by carefully lifting the poke hole stoppers in the top, stopping the blower for a moment while so doing. When the fuel bed reaches this stage, the gas made should be of about the proper quality. To test it, lift one of the stoppers in the top slightly and ignite the gas escaping. If the flame burns red—not blue—while the blower is working full blast, it may be considered satisfactory and may be led to the scrubber, by closing the purge valve. Beyond the scrubber the gas, which may be lighted at any convenient test opening, should, when of good quality, burn with a long steady flame of blue color with a distinct reddish tinge. If, on the other hand, the flame shows faint blue together with a tendency to go out, the gas is of poor quality and mistakes have been made somewhere in the process.

During further operations more fuel may be added as long as the upper layer always turns a dull red. The proper mean height or thickness of fuel bed should be about 30 in.—for hard coal a little less, for coke a little more. To obtain a uniform quality of gas, the level of the bed should be maintained at about the same point. The location of the level may be determined at several places by inserting a rod through the poke holes, the covers of which may be slightly lifted for the purpose.

At least once in a day of 10 hours the doors at the bottom of the producer must be opened, after first opening the purge valve and shutting off the blast. *This operation must always be carried out in that order;* open first the purge valve, next stop the blower. The ash may then be removed and the grate freed from clinkers. This work should be carefully done in order to preserve the producer lining as long as possible. At the same time the proper carrying out of this cleaning process is of importance as far as the satisfactory operation of the plant is concerned. During the shut-down at the noon hour, open the purge valve partly and stop the blast, so that the fire can get sufficient draft to keep it alive. The blast should be started again about 15 minutes before work is resumed, if a good quality of gas is desired at that time.

After work is stopped at night, draw the fire in the boiler, increase the thickness of the fuel bed in the producer to about 3 ft., and open the purge valve. Under these conditions the fire in the generator will keep alive until morning. In the morning the burned down fuel column may, with the aid of the blast, be again brought to normal height in from 30 to 45 minutes. The best time to draw out the ashes is in the morning just before starting the blast.

During the time that the producer is in operation, the top should be water-cooled to keep it from burning out too rapidly, and the water should also not be shut off over night.

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[1] See illustration, Fig. 406.

From time to time, depending upon the care that the producer receives, the charge should be completely drawn to free the lining from adhering clinkers.

3. **Scrubber and Piping.** The scrubber should be opened about every eight days to remove the mud deposited. Special attention should here be paid to the syphon and the overflow pipe. When the water is at its normal level, the syphon should dip into the water about .8 in. The tubes of the air pre-heater should from time to time be freed from soot, taking off the cover plates and using wire brushes to scrape the tubes.

If, on account of fouling, the piping offers greater resistance to the flow of gas, the quality of the latter will deteriorate.

4. **Sawdust Purifier.** Where the best quality of hard coal is used, no sawdust purifier is necessary. For tarry fuels or those carrying dust, however, the use of such purifiers is recommended. The spaces between the wooden grids or trays should be filled with dry sawdust packed as little as possible. About once in two weeks the filling should be renewed. If the old charge is thoroughly dried it may be used over again. To prevent the sawdust from falling through the grids, a layer of short planing chips may first be put down. After the cover of the purifier has been fastened down, a quantity of gas is allowed to escape, until it is certain that all the air has been driven out.

5. **Gas Holder.** The bell of the holder is employed to regulate the quantity of gas made.

The gas pressure is easily regulated by adjusting the counterweights. Care should be taken not to work with too high a pressure, as this interferes with gasification. In general, the pressure should not exceed about 3 in. of water. Periodically, about twice a day, the water collectors in the pipe line of the gas holder and those at the low points in the gas mains should be pumped out by means of the pump furnished for this purpose.

6. **In General.** When a plant is first started up, or after any cleaning operations, a sufficient quantity of gas should be allowed to escape to the open air through the ventilating cocks or valves placed at several points. The purpose of this is to replace any air in the apparatus, because air forms an explosive mixture with the gas. It is also recommended, before opening scrubber or purifier for cleaning purposes, to blow air through the system by means of the steam blast, the producer being cold, to make sure that the laborers will not breathe the gas.

Any leaks in any of the apparatus or piping should be immediately taken care of, as the gas is poisonous.

These instructions are not to be considered as imposing responsibility of any kind upon Körting Bros.

6. Instructions for the Operation of Suction-Gas Producers made by the Gasmotoren-Fabrik Deutz[1]

1. **Method of Operation.** *General Directions.* The engine draws air saturated with water vapor through the producer filled with incandescent coal or coke and through the scrubber and gas tank. The air and steam passing through the incandescent fuel columns are transformed into the gas which serves to drive the engine.

The air takes up the necessary water vapor by leading it over the surface of water in the vaporizer built around the top of the producer and heated by the gases as they pass out of the producer. The water level in the vaporizer is maintained at a certain point by supplying a constant stream of water, and allowing the excess to flow through a small overflow pipe into the air pipe and from here into the ash-pit where it evaporates and helps to keep the grate cool.

During operation, on account of the suction of the piston, apparatus and piping are under a pressure less than atmospheric, so that if any leak occurs or any valve is opened, air will flow into the system but no gas will leak out. On this account the gas will, under such circumstances, be diluted, decreasing the capacity of the engine. If the quantity of air leaking in is considerable, there is also the danger that an explosive mixture may be formed.

It is therefore of prime importance to have all piping, valves and machinery gas tight.

The best means to test the tightness of the entire system is by means of the blower used to start the producer. If air is blown into the system each joint may be quickly tested by means of a flame. This examination should be undertaken not only after the plant has first been erected, but at stated periods also during regular operation. Air may, however, also find its way into the piping by improperly handling the valves. *These should therefore be handled only as laid down in the following directions, and it is important to perform the operations only in the sequence indicated.*

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[1] See illustration, Fig. 412.

The producer room should always be provided with *proper ventilation.* The coal used should be screened over a sieve having about 6 or 8 meshes to the inch, in order to remove the fine dust.

2. **Starting the Producer, after Erection or after it has been Cleaned.** Before starting, the entire plant should be inspected and the engine made ready for starting. Special care should be taken to see that the water seal at the scrubber is filled, that there is water in the ash pit, and that the vaporizer is full to the overflow. Close the scrubber valve and the water vapor valve and open the blower valve, and the ash and fire-doors. Next pull the removable chimney down onto its seat at the top and start a wood fire in the generator as in a common stove. After the wood is well lighted add coal, and after this is burned through keep on adding it slowly until the producer is filled.

In order to bring the charge to incandescence more rapidly the blower is brought into operation just after the fire is well started, or at least when the generator is filled. Ash and fire-doors must of course first be closed and the top cover put in place. The blower must be kept going until the gas burns at the try cock with a steady flame. Next the grate should receive a cleaning, for which purpose the blower is stopped, the blower valve is closed, the ash pit and fire-doors are opened. The clinkers are next loosened by poking from below through the grate and are then removed through the fire-door without drawing any incandescent coal. By poking upwards through the fire-doors any clinker adhering to the walls above the doors may also be loosened and taken out. During this period the quality of the gas has of course been falling off, for which reason, after closing the lower doors and opening the blow valve, the blower is again put on until the gas again proves of good quality.

When the plant is first started, and also when for some reason there is no gas left from the last operation in scrubber or piping, gas must be forced through the system by means of the blower before the engine can start. To this end open the purge cock leading to the outer air at the engine, open the scrubber valve and close the filling hopper. Gas must then be blown through until the air is entirely displaced and the gas burns steadily just ahead of the engine.

3. **Starting the Engine.** After making certain that the gas is of good quality at the test cock, the engine may be started. For this purpose stop the blower, open the scrubber valve and the water vapor valve, close the charging hopper, and start the water supply to the scrubber Then start the engine, closing the purge valve at the same time.

Between the stopping of the blower and the starting of the engine, no time must be lost. All arrangements required for starting the engine should therefore be made beforehand.

During operation the water supply to the vaporizer should be such that the overflow is sufficient to maintain a pool of water in the ash pit.

4. **Charging the Producer.** The level of the coal should not be allowed to sink below the lower edge of the charging funnel. When charging during operation becomes necessary, it must be done immediately after lifting the cover, and the latter must be replaced immediately thereafter.

The hopper or funnel, either during operation or during stand-by periods, should not be open for any length of time without putting the chimney in place.

5. **Stopping Operation.** Stop the engine by following the directions given under the corresponding head in the engine instructions, open the purge valve at the engine and also the charging hopper. Next close the scrubber valve and the purge valve. The closing of these two valves must not be forgotten in order to retain a good quality of gas in the system beyond the generator to aid in the next starting operation. Stop the water supply to the scrubber and to the vaporizer. Open the air valve so that the producer can get enough air to keep the fire alive and close the air-steam pipe.

6. **Starting after Short Interruption.** Starting the producer, the contents of which are still incandescent, may be done as follows: Open the fire and ash pit doors, loosen the clinkers with a poking bar nad remove them, carrying away as little coal as possible. Next close the doors again, open the blower valve and blow air through the producer until the gas again burns properly at the test cock. After this proceed as directed under 2 and 3.

7. **Cleaning the Grate.** Depending upon the kind of coal used, it becomes necessary to clean the grate at intervals of several hours. *This may be done during operation* by opening the ash

pit doors and raking the grate from the center to front and back, thus pushing the clinkers in front of the fire-doors. If clinkers also form on the sides of the producer above the fire-doors, these can only be removed by poking upward through these doors. But the fire-doors must remain open only a very short time and the operation must therefore be performed very quickly.

If the generator is badly clinkered up, the only remedy is to draw the fire altogether and to clean thoroughly, the clinkers at the sides being removed by poking through the charging hopper. This operation had best be undertaken every Monday morning. It is recommended not to clean in this manner just after shutting down on Saturday, because the producer would cool down too far, and the lining may suffer under the frequent extreme temperature changes.

8. **Cleaning the Piping.** The piping should be examined and cleaned once a month. It should be noted that the fouling takes place mostly in the bends and elbows and at the joints of the piping to the scrubber. To clean the latter places, the flange unions must be disconnected.

9. **The Scrubber.** The scrubber is filled with coke of such size that the pieces can not fall through the grate at the bottom. The water is supplied at the top and wets the coke on its way down. The supply of water must be so regulated that the gas leaves the scrubber completely cooled down.

The scrubber should be cleaned once in from 9 to 12 months, depending upon the hours of operation per day.

To accomplish this the plant must be put out of commission. Close the scrubber valve and take off the upper and lower covers of the scrubber. *After this let the scrubber stand for several hours in order to let the gas escape.* Next, draw out the coke by means of an iron rake, through the lower door, and put in the new charge. Finally, replace the doors or covers, replacing any defective packing.

All cleaning operations should be carried on only in the day time, and no fire, open flame, or smoking should be allowed in the room during that time. The work should be done by at least two men and ample supply of fresh air should be provided for.

It is urgently recommended that, independent of any general directions that may be given, there should be posted somewhere in the producer room a brief, easily memorized special notice, which may read about as follows:

7. Instructions for the Prevention of Accidents in the Operation of Suction-Gas Plants

(Güldner-Motoren Gesellschaft, Aschaffenburg.)

1. The outer cover of the charging hopper must not be lifted unless the inside conical cover is pressed against its seat by the counterweight. The simultaneous opening of both covers is dangerous and therefore absolutely prohibited.

2. While charging coal into the upper hopper, or removing clinker by poking through the openings provided, the attendant must hold his face to one side far enough so that any flame shooting out can not reach it. Another reason for this rule is that it avoids the breathing of any gas that may be present in the charging hopper.

3. The dismantling or taking apart of any of the apparatus of a producer plant that has already been in operation, should be undertaken only when the fire in the producer is completely dead, and after the system has been thoroughly aired out. In work of this kind the precaution to have no open flame whatever, by means of which the gas could be ignited anywhere near the plant, should be strictly observed.

4. The water seals of the scrubber must always be kept filled with water. The scrubber water must be led away only in closed syphons and mains.

II. Specifications, Etc.

1. General Specifications for the Purchasing of Machinery

(Adopted by the "Verein Deutscher Maschinen-Fabriken" at Cologne, 28th Dec., 1889, and 10th Jan., 1891, and at Hamburg, 27th May, 1893.)

1. Prices are free on board works; packing and freight are subject to special agreement.

2. Payment must be made in legal tender at the offices of the manufacturer (or selling agent) and at the following rates: one-third of the purchase price at the time of placing the order, one-third after the main parts are delivered f.o.b. works, and one-third three months after the machine was first put in service. In any case, however, the last payment must be made in six months from the second payment, if delay in starting the machine occurs through no fault of the manufacturer or selling agent. Payment by monthly installments is permissible, but in such case the average monthly rate should be so fixed as to cancel the obligation in about the time outlined above.

3. The seller *guarantees* construction and workmanship for a period of ..... months, agreeing to replace without charge during that time all parts that prove defective or useless on account of inferior material, poor design, or bad workmanship, or to remedy all defects and troubles chargeable to the same causes.

Natural wear and tear is of course not covered by this guarantee.

4. *Delay* in delivery for which the seller is responsible entitles the purchaser to a deduction of not to exceed ½ of 1% of the purchase price for every full week that delivery is delayed.

5. Any other claims for damages than those defined in paragraphs 3 and 4 are not recognized.

6. Strikes abrogate any agreements concerning time of delivery.

7. Before beginning with erection, the foundations, etc., must be furnished and completely settled and the place of erection must be protected against stress of weather.

8. The manufacturer furnishes one or more skilled erectors, as may be agreed upon. The purchaser supplies the erectors free of cost, with the necessary help, hoisting facilities, light, etc., and also with minor materials like oil, grease, red lead, cotton waste, etc. The helpers remain in the employ of the purchaser.

9. The charges for each erector, besides traveling fare, shall be as follows: \$..... for each day traveled, and \$..... per working hour, also \$..... per day and per man for board and lodging. If required, the manufacturers or seller may guarantee that the total costs for one or more erectors shall not exceed a certain fixed sum. Ten hours is considered a *working day*, except in mines, etc. ("unter Tage"), where eight hours are considered equivalent to ten hours. Overtime and labor on Sunday are charged for as per special agreement.

10. *Foundation plans* are furnished in each case without cost, and, where required, also a *list of parts*.

11. Any disagreements regarding the interpretation and fulfilment of the contract shall be referred to a *board of arbitration*, to which each party appoints one member. These two then agree upon a third member before commencing negotiations. Should the two representatives of the parties concerned fail to reach an agreement, each presents a report to the third member of the board, whose decision is final. The costs of the process are divided between the parties as agreed upon by the board or decided upon by the third member.

III. Regulations Concerning the Installation and Use of Internal-Combustion Engines

Under this head the German text points out that the installation of gas producers in Germany does not in general require government license unless the plant includes a closed boiler used under pressure. In this case permission for the installation of the boiler must be obtained and it is subject to inspection under government rules and regulation. In some localities, however, the authorities class both pressure and suction producers under the general head of "furnaces," in which case their installation is subject to the general building laws.

The installation of oil engines is in general not subject to government regulations. In some cases, however, insurance companies prescribe certain measures in the interest of safety.

As far as the translator's knowledge goes, there is at present in the United States no government supervision in the case of either suction or pressure gas producers. Insurance companies concern themselves with oil engines, gasoline, and perhaps crude oil, but not kerosene engines, only in so far as to lay down in the insurance policies for buildings certain regulations concerning the handling and storing of the oil. For these reasons the government and insurance regulations translated rather fully in the following subdivisions may be of little interest to the general reader, but they will serve to show what points it was thought necessary to cover to make the operation of producer gas plants and oil engines safe to both machinery and attendants.

It is further pointed out in the text that the restrictions put upon the use of the lighter hydro-carbon oils, like gasoline, are too strict, and that the danger of explosions is much exaggerated. That the latter is in general the case may be seen from the following extract from a report by Dr. Leymann, Wiesbaden:

> "On account of many small fires due to gasoline, kerosene, etc., the idea that these liquids, as well as the similar lighter coal-tar oils, like benzol, are explosive in themselves, has gained wide acceptance. From extensive information received it appears, however, that although these substances are highly inflammable and easily fired, they are not in themselves explosive. This fact may also be easily seen by considering their chemical composition, the main constituents being carbon and hydrogen. No one has as yet succeeded in firing these compounds either through jar or impact, and where explosions have occurred the presence of mixtures consisting of air with the vapors from these materials has always been proven. The vapors alone cannot be ignited, they may be led through incandescent tubes with safety. Even the air-vapor mixtures are explosive only under certain stated conditions and within very narrow limits. The more volatile the material, the greater the danger of the formation of explosive mixtures, benzine and gasoline heading the list in this respect. The following mixture ratios of vapor to air are explosive: for benzol, upper limit $\frac{1}{35}$, lower limit $\frac{1}{14}$; for gasoline, upper limit $\frac{1}{41}$, lower limit, $\frac{1}{19}$, that is, there must be present in the mixtures per unit volume of gasoline vapor not less than 19 and not more than 41 unit volumes of air. All other mixtures of the same vapor with air simply burn without explosion, and only those are explosive in which we have, expressing the above ratios in per cent, from 2.5 to 6.7% of benzol, or from 2.3 to 4.8% of gasoline vapor. The probability that an explosive mixture may be formed is consequently quite small. Concerning closed vessels completely filled with gasoline, therefore, these can explode only as a steam boiler explodes, that is, through excess of pressure. It is possible, however, although this has been known to occur only once, for an explosion to take place when the vessel is not entirely filled, in which case the vapor may form an explosive mixture with the air contained. The actual cause of the explosion in the one case cited could not be ascertained. The accident was quite likely due to the same causes that are at the bottom of explosions in air compressors. Vessels holding benzine and similar liquids need therefore be considered dangerous only in case the liquids are in contact with compressed air. Conflagrations, even when large quantities of these volatile fuels were concerned, which is, however, rarely the case, have been accompanied by explosions, but the flames are in general easily controlled by earth embankments and walls. Store-rooms for gasoline may advantageously be enclosed by embankments of that kind and every large tank or barrel should be similarly isolated. Since it is hardly likely to avoid spilling some of the liquid, through careless handling or otherwise, provision for instant drainage should be made. Finally, all gasoline store-rooms, or rooms in which gasoline is used to any extent, should be furnished with means of rapidly filling the room with steam, as this will quickly and thoroughly smother any fire. The steam pipe serving for this purpose must be provided with a valve outside of the room or building.

1. Regulations for the Installation and Operation of Suction-Gas Power Plants

(Issued by the Prussian Secretary for Commerce and Industry, June 20, 1904.)

1. The apparatus for the production and cleaning of suction gas and the gas engine must be installed in well lighted rooms at least 11.5 ft. and in the case of engines over 50 H.P., at least 13 ft. high. These rooms must have ample ventilation in such manner that the accumulation of gas cannot occur. The rooms must not be used for other purposes. It is permissible to install the entire plant in a single room.

2. The use of cellar rooms is only permissible when the floor of such cellars is not to exceed 6.5 ft. below the level of the ground outside.

3. A direct communication of rooms used for such purposes with living rooms is not allowed. It is also compulsory to prevent the access of hot air or vapors to living rooms or work shops adjacent to or above the engine or producer room.

4. The floor space of the plant must be of such size that the individual apparatus, engine and other accessories may be easily and safely accessible from all sides. Especial care should be had to see that pipes do not hinder freedom of movement or interfere in any way with proper attendance to producers and engines.

5. The charging of the producers must be possible without danger to the attendant (from fixed platforms, stairs, or ladders). Care should be taken to see that no gas can escape through the charge opening into the room.

6. The gases produced in the generator during starting up or during stand-by periods must be conducted by a gas-tight pipe of proper size above the eaves of the neighboring buildings. The exhaust gases from the engine should be carried through a separate iron pipe to the same level and in such a way that the neighborhood is not annoyed by the noise.

7. Arrangements must be made whereby the escape of gas from the producer into the rooms during starting up and stand-by periods is prevented.

8. Arrangements must also be made whereby, in case of mis-fires or other troubles, the explosive mixture is prevented from flowing back into the gas mains, and explosions of such mixtures in the exhaust main are rendered impossible.

9. Care should also be taken to reduce to a minimum the undesirable features accompanying the cleaning of the producer (drawing of ash, removing clinker, etc.). It may be necessary in some cases to collect and lead away the hot vapors and gases forming.

10. Gas washing and scrubbing apparatus should be furnished with instruments (gauges, manometers) to indicate the existing pressure.

11. The waste water discharged from the washing apparatus must be so treated that it leaves the plant without odor and completely neutralized. Ashes and other refuse also must be so taken care of as to avoid all annoyance to the neighborhood.

12. The ventilating arrangements must not annoy the attendants through drafts or the neighborhood through noise or other causes.

13. If the radiant heat from the gas producers is likely to cause trouble, the producers must be suitably covered. Exhaust pipes inside the rooms or buildings must also be either cooled or covered.

14. The foundations for engines and other machines in the plant must be so built as to prevent annoying the neighborhood with vibrations.

15. Provision must be made for suitable artificial lighting.

16. The provisions of the law for the prevention of accidents must be carried out.

17. The attendants must be furnished with seats and with washing facilities.

18. In case the gas plant is combined with a storage-battery installation, the rooms for the latter must be separate from the former, and must be properly ventilated. The artificial lighting of the battery rooms must be furnished only by incandescent lamps with special safety hood or by lights from the outside.

19. The preceding regulations are not to interfere with any rules or regulations already established by local building or other laws.

2. Regulations for the Installation and use of Gasoline Engines for Agricultural Purposes[1]

(Issued by the Society of German Fire Insurance Companies.)

I. Stationary Engines

A. Regulation Covering Gasoline Engines, the Vaporizer for which is Installed in a Separate Room

1. The engine must be installed in a room devoid of artificial heat and separated from other rooms by walls without openings other than those necessary to pass shafting, etc. There must be no combustible materials either stored or handled in the room, and the engine itself must be erected on a fire-proof foundation. In cases where this foundation does not extend about 12 in. all around the base of the engine, any wooden floor must be covered with sheet iron to at least that distance from the engine base. Above the engine no woodwork or combustible should be tolerated within 3 ft. of the engine; at the sides this clearance should be at least 18 in.

2. The exhaust pipe near the engine must be covered by some fire-proof material to remove danger of fire from this source.

3. The room containing the gasoline vapor producer must not communicate with any other room, except through the necessary wall openings for the passage of pipes. The walls must be heavy, and both floor and ceiling should be stone or concrete. Good ventilation must be provided and the room must not be heated.

4. In case artificial lighting is necessary for the generator room, the light must either be furnished from the outside, or incandescent lamps or safety lamps built on the Davy principle must be used.

5. The charging of the generator must be done only in the day time and from a steel tank located in the room itself, which tank should not have a maximum capacity exceeding 200 kg. (about 75 gallons) of gasoline. The liquid must flow only through closed pipes and must be moved only by enclosed rotary pumps.

6. No other gasoline except that permitted under Par. 5 may be kept on the property insured.

B. Regulations for Gasoline Engine Installations in which the Vaporizer is Installed in the Same Room

1. Engine and vaporizer may be installed only in a separate room containing no combustible material. The room must be properly ventilated and must not be heated. The room must be separated from adjacent rooms by heavy walls without openings and must have fire-proof floor and ceiling. Not even passages for belts or rope drives are permitted in the walls separating the engine room from the others, but shafting may be allowed to pass through, provided it is closely surrounded at the wall with fire-proof material.

2. The engine and vapor generator room can be artificially lighted only by light transmitted from without, or by electric glow lamps whose closely fitting globes must also cover the lamp sockets, or by safety lamps constructed on Davy's principle. Any other handling of light or fire in this room is absolutely prohibited.

3. To operate the engine only electric ignition must be used.

4. See Par. 2 under *A*.

5. See Par. 5 under *A*.

6. See Par. 6 under *A*.

C. Regulations Covering Gasoline Engines which do not Employ Vaporizers

1. See Par. 1 under *A*.

2. The gasoline tank belonging to the engine (that is the tank containing the supply for the engine and for the ignition flame) must be located outside of the engine room in a separate

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[1] These regulations also apply to those engines which use ligroin, naphtha, or any other oil whose flash-point is below that of kerosene.

room neither artificially lighted or heated, and in which any handling of light or fire is strictly prohibited. The supply room must have heavy side walls, fire-proof floor and ceiling, and the walls can only be broken through for the passage of the small supply of pipes leading to the engine.

3. See Par. 2 under *A*.
4. If alcohol is used for the ignition flame, only the quantity required for daily consumption may be kept in the engine room and it must be stored in a tightly closed sheet iron or steel tank.
5. Concerning the filling of the gasoline supply tanks, see Par. 5 under A.
6. See Par. 6 under *A*.

II. Portable Engines

A. Regulations Covering Gasoline Engines with Separate Vapor Producer

1. Portable engines (locomobiles) must be placed at least 15 ft. from any buildings, barns, sheds, and threshing machines, and no easily combustible material must be allowed to get any nearer than this distance.
2. Only electric ignition must be used in operating the engine.
3. See Par. 2 under **I**, *A*.
4. The amount of gasoline in storage must not exceed 200 kg. (about 75 gallons) and must be kept in a sheet steel tank. The place of storage should be either a substantial room or building with fire-proof floors and ceilings, well ventilated but not heated, and having no communication with other rooms; or a pit in the ground covered with an iron plate and located at least 100 ft. from any buildings or combustible material.
5. The engine vaporizer must be filled only in the day time direct from the storage tank by means of closed ducts or pipes and enclosed rotary pump. For this purpose the engine must always be moved to the place of storage.

   The vaporizer must also be emptied in the day time and only through a screw connection with the storage tank mentioned under Par. 4 above.
6. The engine can only be placed in its shelter on the property after the vaporizer has been completely drained.

B. Regulations Covering Gasoline Engines which do not Employ Vaporizers

1. See Par. 1 under **II**, *A*.
2. See Par. 2 under **II**, *A*.
3. See Par. 3 under **II**, *A*.
4. See Par. 4 under **II**, *A*.
5. See Par. 5 under **II**, *A*, except that "gasoline tank" should be substituted for "vaporizer."
6. See Par. 6 under **II**, *A*, except that "gasoline tank" should be substituted for "vaporizer."

3. Special Regulations Covering the Use of Portable Kerosene Engines in the Agricultural Industries

(Issued by the Society of German Fire Insurance Companies.)

1. See Par. 1 under **2**, **II**, *A* above.
2. The fuel used must be only common kerosene having a specific gravity of at least .80.
3. The kerosene tanks may be filled only in the day time, and no kerosene or alcohol supplies should be stored in the immediate vicinity (see under **2**, **II**, *A*).

4. Special Regulations Covering the Use of Alcohol Engines in Agricultural Industries, as well as for General Industrial Purposes

(Issued by the Society of German Fire Insurance Companies.)

1. The room in which the engine is installed must be separated from other rooms by thick walls without openings (except those for the transmission), it must have fire-proof floor and ceiling, must not be heated or used for any other purpose, and should not contain any combustible materials.

2. No lamps of any kind, except incandescent lamps, must be brought any nearer than 7 ft. to the engine or any tank or vessel containing alcohol.

For general industrial purposes, paragraphs 1 and 2 above show a somewhat different form:

1. The engine must be installed only in a room in which there are no easily inflammable materials either stored or handled, and must be placed on a fireproof foundation. If this foundation does not project beyond the engine base at least 12 in. on all sides, any wooden floor must be covered to that distance with sheet iron. Above engine and alcohol tanks woodwork or other combustible materials must be kept at a distance of at least 3 ft. while at all sides the clearance should be at least 18 in.

2. Both engine and vessels containing alcohol must be distant at least 7 ft. from any heated stove or heated pipes, and no illuminating agents, except incandescent lamps, should be brought any nearer than that distance.

3. The exhaust pipe near the engine must be covered by some fire-proof material.

4. The alcohol supply tank must not be placed *directly over* the engine, but should be securely fastened at a distance of at least 7 ft. to one side. The tank or tanks must be made of sheet metal, and must be furnished with tight covers. The alcohol must flow from the tank to the engine only in closed ducts or pipes.

5. The supply tank should be filled only in the day time whenever that is possible, but where artificial illumination is necessary only incandescent lights or safety lamps must be used.

6. The only supply of alcohol allowed in the engine room is that required for a single filling of the supply tank. It must be kept in tight sheet metal cans and even then should only be brought in just before it is needed.

7. The supply of alcohol in storage must not exceed 250 kg. (=about 80 gallons). It must be kept in a fire-proof chamber or vault or outside of the buildings insured in a place such that these buildings can not be endangered.

IV. Regulations Concerning the Testing of Gas Engines and Gas Producers

1. Rules for Conducting Tests of Gas and Oil Engine

(Am. Soc. of Mech. Engineers, Code of 1901.)

1. **Objects of the Tests.** At the outset the specific object of the test should be ascertained, whether it be to determine the fulfilment of a contract guarantee, to ascertain the highest economy obtainable, to find the working economy and the defects as they exist, to ascertain the performance under special conditions, or to determine the effect of changes in the conditions; and the test should be arranged accordingly.

Much depends upon the local conditions as to what preparations should be made for a test, and this must be determined largely by the good sense, tact, judgment, and ingenuity of the expert undertaking it, keeping in mind the main issue, which is to obtain accurate and reliable data. In deciding questions of contract, a clear understanding in regard to the methods of test should be agreed upon beforehand with all parties, unless these are distinctly provided for in the contract.

2. **General Condition of the Engine.** Examine the engine, and make notes of its general condition, and any points of design, construction, or operation which bear on the objects in view. Make a special examination of all the valves by inspecting the seats and bearing surfaces, and note their condition, and see if the piston rings are gas-tight.

If the trial is made to determine the highest efficiency, and the examination shows evidence of leakage, the valves and piston rings, etc., should be made tight, and all parts of the engine put in the best possible working condition before starting on the test.

3. **Dimensions, etc.** Take the dimensions of the cylinder, or cylinders, whether already known or not; this should be done when they are hot, and in working order. If they are slightly worn the average diameter should be determined. Measure, also, the compression space for clearance volume, which should be done, if practicable, by filling the spaces with water previously measured, the proper correction being made for the temperature.

4. **Fuel.** Decide upon the gas or oil to be used, and if the trial is to be made for maximum efficiency, the fuel should be the best of its class that can readily be obtained, or one that shows the highest calorific power.

5. Calibration of Instruments used in the Tests. All instruments and apparatus should be calibrated and their reliability and accuracy verified by comparison with recognized standards. Apparatus liable to change or become broken during the tests, such as gauges, indicator springs, and thermometers, should be calibrated both before and after the experiments. The accuracy of all scales should be verified by standard weights. In the case of gas or water meters, special attention should be given to their calibration, both before and after the trial, and at the same rate of flow and pressure as exists during the trial.

(*a*) *Gauges.* For pressures above the atmosphere, one of the most convenient, and at the same time reliable, standards is the dead-weight testing apparatus which is manufactured by many of the prominent gauge makers. It consists of a vertical plunger nicely fitted into a cylinder containing oil or glycerine, through the medium of which the pressure is transmitted to the gauge. The plunger is surmounted by a circular stand on which weights may be placed, and by means of which any desired pressure can be secured. The total weight, in pounds, on the plunger at any time, divided by the area of the plunger and of the bushing which receives it, in square inches, gives the pressure in pounds per square inch.

Another standard of comparison for pressures is the mercury column. If this instrument is used, assurance must be had that it is properly graduated with reference to the ever varying zero point; that the mercury is pure, and that the proper correction is made for any difference of temperature that exists, compared with the temperature at which the instrument was graduated.

For pressure below the atmosphere, an air pump or some other means of producing a vacuum is required, and reference must be made to a mercury gauge. Such a gauge may be a U-tube having a length of 30 in. or so, with both arms properly filled with pure mercury.

(*b*) *Thermometers.* Standard thermometers are those which indicate 212° F. in steam escaping from boiling water at the normal barometrical pressure of 29.92 in., the whole stem up to the 212° point being surrounded by the steam; and which indicate 32° F. in melting ice, the stem being likewise completely immersed to the 32° point; and which are calibrated for points between and beyond these two reference points. We recommend, for temperatures between 212 and 400° F., that the comparison of the thermometer be made with the temperature given in Regnault's steam tables, the method required being to place it in a mercury well surrounded by saturated steam under sufficient pressure to give the right temperature. The pressure should be accurately determined as pointed out in the above section (*a*), and the thermometer should be immersed to the same extent as it is under its working condition.

Thermometers in practice are seldom used with the stems fully immersed; consequently, when they are compared with the standard, the comparison should be made under like conditions, whatever those happen to be.

If pyrometers of any kind are used, they should be compared with a mercury thermometer within its range, and if extreme accuracy is required with an air thermometer, or a standard based thereon, at higher points, care being taken that the medium surrounding the pyrometer, be it air or liquid, is of the same uniform temperature as that surrounding the standard.

(*c*) *Indicator Springs.* The indicator springs should be calibrated with the indicator in as nearly as possible the same condition as to temperature as exists during the trial. This temperature can usually be estimated in any particular case. A simple way of heating the indicator is to subject it to a steam pressure just before calibration. Compressed air, or compressed carbonic acid gas, are suitable for the actual work of calibration. These gases should be used in preference to steam, so as to bring the conditions as near as possible to those which obtain when the indicators are in actual use. When compressed carbonic acid gas is used, and trouble arises from the clogging of the escape valves with ice, the pipes between the valve and the gas tank should be heated. With both air and carbonic acid gas, the pipes leading to the indicator should also be heated if it is found that they are below the required temperature. The springs may be calibrated for this class of engines under a constant pressure, if desired, and the most satisfactory method is to cover the whole range of pressure through which the indicator acts; first, by gradually increasing it from the lowest to the highest point, and then gradually reducing it from the highest to the lowest point, in the manner which has heretofore been widely followed by indicator makers; a mean of the results should be taken. The calibration should be made for at least five points, two of these being for the pressures corresponding to the maximum and minimum pressures, and three for intermediate points equally distant.

The standard of comparison recommended is the dead weight testing apparatus, a mercury column, or a steam gauge, which has been proved correct by reference to either of these standards.

When the scale of the spring determined by calibration is found to vary from the nominal scale with substantial uniformity, it is usually sufficiently accurate to take the arithmetical mean of the scales found at the different pressures tried. When, however, the scale varies considerably at the different points, and absolute accuracy is desired, the method to be pursued is as follows: Select a sample diagram and divide it into a number of parts by means of lines parallel to the atmospheric line, the number of lines being equal to and corresponding with the number of points at which the calibration of the spring is made. Take the mean scale of the spring for each division and multiply it by the area of the diagram inclosed between two contiguous lines. Add all the products together and divide by the area of the whole diagram; the result will be the average scale of the spring to be used. If the sample diagram selected is a fair representative of the entire set of diagrams taken during the test, this average scale can be applied to the whole. If not, a sufficient number of samples of diagrams representing the various conditions can be selected, and the average scale determined by a similar method for each, and thereby the average for the whole run.

(*d*) *Gas Meters.* A meter used for measuring gas for a gas engine should be calibrated by referring its readings to the displacement of a gasometer of known volume, by comparing it with a standard gas meter of known error, or by passing air through the meter from a tank in which air under pressure is stored. If the latter method is adopted, it is necessary to observe the pressure of the air in the tank and its temperature, both at the tank and at the meter, and this should be done at uniform intervals during the progress of the calibration. The amount of air passing through the meter is computed from the volume of the tank and the observed temperatures and pressures.

The volume of gas thus ascertained should be reduced to the equivalent at a given temperature and atmospheric pressure, corrected for the effect of moisture in the gas, which is ordinarily at the saturation point or nearly so. We recommend that a standard be adopted for gas-engine work, the same as that used in photometry, namely, the equivalent volume of the gas when saturated with moisture at the normal atmospheric pressure at a temperature of 60° F. In order to reduce the reading of the volume containing moist gas at any other temperature to this standard, multiply by the factor

$$\frac{459.4+60}{459.4+t} \times \frac{b-(29.92-s)}{29.4},$$

in which b is the height of the barometer in inches at 32° F., t the temperature of the gas at the meter in degrees F., and s the vacuum in inches of mercury corresponding to the temperature of t obtained from steam tables.

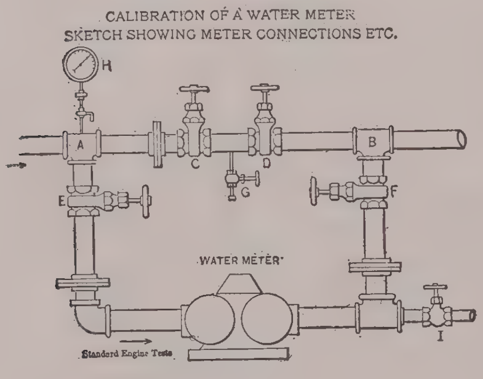
CALIBRATION OF A WATER METER
SKETCH SHOWING METER CONNECTIONS ETC.
H
A
B
C
D
G
E
F
WATER METER
I
Standard Engine Tests

Fig. 725.

(*e*) *Water Meters.* A good method of calibrating a water meter is the following, reference being made to Fig. 725.

Two tees A and B are placed in the feed pipe, and between them two valves C and D. The meter is connected between the outlets of the tees A and B. The valves E and F are placed one on each side of the meter. When the meter is running, the vavles E and F are opened, and the valves C and D are closed. Should an accident happen to the meter during the test, the valves E and F may be closed, and the valves C and D opened, so as to allow the feed water to flow directly to the point of use. A small bleeder G is opened when the valves C and D are closed, in order to make sure that there is no leakage. A gauge is attached at H. When the meter is tested, the valves C, D, and F are closed, and the valves E and I are opened. The water flows from the valve I to a tank placed on weighing scales. In testing the meter the rate of flow should be the same as that on the test, and the water leaving the meter is throttled at the valve I until the pressure shown by the gauge H is the same as that indicated when the meter is running under the normal conditions. The piping leading from the valve *I* to the tank is arranged with a swinging joint, consisting merely of a loosely fitting elbow, so that it can be swung readily into the tank or away from it. After the desired pressure and rate of flow have been secured, the end of the pipe is swung into the tank the instant that the pointer of the meter is opposite some graduation mark on the dial, and the water continues to empty into the tank. The tests should be made by starting and stopping at the same graduation mark on the meter dial, and continued until at least 10 or 20 cu.ft. are discharged for one test. The water collected in the tank is then weighed.

The water passing the meter should always be under pressure in order that any air in the meter may be discharged through the vents provided for this purpose. Care should be taken that there is no air contained in the water. The meter should be tested both before and after the engine trial, and several tests be made of the meter in each case in order to obtain confirmative results. It is well to make preliminary tests to determine whether the meter works satisfactorily before connecting it up for an engine trial. The results should agree with each other for two widely different rates of flow.

6. **Duration of Test.** The duration of a test should depend upon its character and the objects in view, and in any case the test should be continued until the consecutive readings of the rates at which oil or gas is consumed, taken at say half-hourly intervals, become uniform and thus verify each other. If the object is to determine the working economy, and the period of time during which the engine is usually in motion is some part of twenty-four hours, the duration of the test should be fixed for this number of hours. If the engine is one using coal for generating gas, the test should cover a long enough period to determine with accuracy the coal used in the gas producer; such a test should be of at least twenty-four hours' duration, and in most cases it should extend over several days.

7. **Starting and Stopping a Test.** In a test for determining the maximum economy of an engine, it should first be run a sufficient time to bring all the conditions to a normal and constant state. Then the regular observations of the test should begin, and continue for the allotted time.

If a test is made to determine the performance under working conditions, the test should begin as soon as the regular preparations have been made for starting the engine in practical work, and the measurements should then commence and be continued until the close of the period covered by the day's work.

8. **Measurement of Fuel.** If the fuel used is coal furnished to a gas producer, the same methods apply for determining the consumption as are used in steam boiler tests.

If the fuel used be gas, the only practical method of measurement is the use of a meter through which the gas is passed. Gas bags should be placed between the meter and the engine to diminish the variation of pressure, and these should be of a size proportionate to the quantity used. When a meter is employed to measure the air used by an engine, a receiver with a flexible diaphragm should be placed between the engine and the meter. The temperature and pressure of the gas should be measured, as also the barometric pressure and temperature of the atmosphere, and the quantity of gas should be determined by reference to the calibration of the meter, taking into account the temperature and pressure of the gas. [See Section 5 (*d*)].

If the fuel is oil, this can be drawn from a tank which is filled to the original level at the end of the test, the amount of oil required for so doing being weighed; or, for a small engine, the oil may be drawn from a calibrated vessel such as a vertical pipe.

In an engine using an igniting flame the gas or oil required for it should be included in that of the main supply, but the amount so used should be stated separately, if possible.

9. **Measurement of Heat-Units Consumed by the Engine.** The number of heat units used is found by multiplying the number of pounds of coal or oil or the cubic feet of gas consumed, by the total heat of combustion of the fuel as determined by a calorimeter test. In determining the total heat of combustion no deduction is made for the latent heat of the water vapor in the products of combustion. There is a difference of opinion on the propriety of using this higher heating value, and for purposes of comparison care must be taken to note whether this or the lower value has been used. The calorimeter recommended for determining the heat of combustion is the Mahler, for solid fuels or oil, or the Junker for gases, or some form of calorimeter known to be equally reliable. (See Poole on "The Calorific Power of Fuels").

It is sometimes desirable, also, to have a complete chemical analysis of the oil or gas. The total heat of combustion may be computed, if desired, from the results of the analysis, and should agree well with the calorimeter values.

In using the gas calorimeter, which involves the determination of the volume instead of the weight of the gas, it is important that this should be reduced to the same temperature as that corresponding to the conditions of the engine trial. The formula to be used for making the reduction is that already given in Section 5 (*d*).

For the purpose of making the calorimeter test, if the fuel used is coal for generating gas in a producer, or oil, samples should be taken at the time of the engine trial, and carefully preserved for subsequent determination. If gas is used, it is better to have a gas calorimeter on the spot, samples taken, and the calorimeter test made while the trial is going on.

10. **Measurement of Jacket Water to Cylinder or Cylinders.** The jacket water may be measured by passing it through a water meter or allowing it to flow from a measuring tank before entering the jacket, or by collecting it in tanks on its discharge.

11. **Indicated Horse-Power.** The directions given for determining the indicated horse-power for steam engines apply in all respects to internal combustion engines.

The indicated horse-power should be determined from the average mean effective pressure of diagrams taken at intervals of twenty minutes, and at more frequent intervals if the nature of the test makes this necessary. With variable loads, such as those of engines driving generators for electric railroad work, and of rubber-grinding and rolling-mill engines, the diagrams cannot be taken too often. In cases like the latter, one method of obtaining suitable averages is to take a series of diagrams on the same blank card without unhooking the driving cord, and apply the pencil at successive intervals of ten seconds until two minutes' time or more has elapsed, thereby obtaining a dozen or more indications in the time covered. This tends to insure the determination of a fair average for that period. In taking diagrams for variable loads, as indeed for any load, the pencil should be applied long enough to cover several successive revolutions, so that the variations produced by the action of the governor may be properly recorded. To determine whether the governor is subject to what is called "racing" or "hunting," a variation diagram should be obtained; that is, one in which the pencil is applied a sufficient time to cover a complete cycle of variations. When the governor is found to be working in this manner, the defect should be remedied before proceeding with the test.

NOTE.—When the engine is governed by the hit-and-miss principle the diagrams taken on one card should in any case cover the series of consecutive explosions, and the mean diagram should be used as the basis of calculations.

The most satisfactory driving rig for indicating seems to be some form of well-made pantagraph, with driving cord of fine annealed wire leading to the indicator. The reducing motion, whatever it may be, and the connections to the indicator, should be so perfect as to produce diagrams of equal lengths, and produce a proportionate reduction of the motion of the piston at every point of the stroke, as proved by test.

To test the accuracy of the reducing motion without making special preparations for a thorough examination, it is sufficient to make a comparison between the actual proportion of the stroke covered and the apparent proportion measured on the indicator, and see how they agree. This may be done on a large engine by making the comparison wherever it happens to stop, and repeating the comparison when it has stopped with the piston at some other point of the stroke. With an engine which can be turned over by hand, or where auxiliary power is provided for moving it, the comparison may be made at a number of equidistant points in the stroke. To make the test properly, a diagram should be taken just before stopping, and this will serve as a reference for the measurements taken after stopping. The actual proportion of stroke covered is determined by measuring the distance which the piston has moved and comparing it with the whole length of the stroke, making sure that the slack has all been taken up. To obtain the apparent indication from the diagram, the indicator pencil is moved up and down with the finger so as to make a vertical mark on the diagram, and the distance of this mark from the beginning of the diagram compared to the whole length of the diagram is the proportion desired.

It is necessary, of course, to go through these operations without changing in any way the adjustment of the driving cord of the indicator, or any part of the mechanism that would alter the movements of the indicator.

In the manipulation of the indicator it is important to keep the instrument in clean condition and preserve it in mechanically good order. Ordinary cylinder oil is the best material to use for lubricating the indicator piston for pressures above the atmosphere. It is better to have the piston fit the cylinder rather loosely—so as to get absolute freedom of motion—than to have a mechanically accurate fit. In the latter case, extreme care and frequent cleanings are required to obtain good diagrams. No diagram should be accepted in which there is any appearance of want of freedom in the movement of the mechanism. A ragged or serrated line in the region of the expansion or compression lines is a sure indication that the piston or some part of the mechanism sticks; and when this state of things is revealed the indicator should not be trusted, but the cause be ascertained and a suitable remedy applied. An indicator which is free when subjected to a steady pressure, as it is under a test of the springs for calibration, should be able to produce the same horizontal line, or substantially the same, after pushing the pencil down with the finger, as that traced after pushing the pencil up and subsequently tapping it lightly. When the pencil is moved by the finger, first up and then down, the piston being subjected to the pressure, the movement should appear smooth to the sense of feeling.

The pipe connections for indicating gas and oil engines should be removed as far as possible from the ports and ignition devices, and made preferably in the cylinder head. The pipes should be as short and direct as possible. Avoid the use of long pipes, otherwise explosions of the gas in these connections may occur.

Ordinary indicators suitable for indicating steam engines are much too lightly constructed for gas and oil engines. The pencil mechanism, especially the pencil arm, needs to be very strong to prevent injury by the sudden impact at the instant of the explosion; a special gas-engine indicator is required for satisfactory work, with a small piston and a small spring.

12. **Brake Horse-Power.** The determination of the brake horse-power, which is very desirable, is the same for internal-combustion as for steam engines.

This term applies to the power delivered from the fly-wheel shaft to the engine. It is the power absorbed by a friction brake applied to the rim of the wheel, or to the shaft. A form of brake is preferred that is self-adjusting to a certain extent, so that it will, of itself, tend to maintain a constant resistance at the rim of the wheel. One of the simplest brakes for comparatively small engines, which may be made to embody this principle, consists of a cotton or hemp rope, or a number of ropes, encircling the wheel, arranged with weighing scales or other means for showing the strain. An ordinary band brake may also be constructed so as to embody the principle. The wheel should be provided with interior flanges for holding water used for keeping the rim cool.

A self-adjusting rope brake is illustrated in Fig. 726, where it will be seen that, if the friction at the rim of the wheel increases, it will lift the weight *A*, which action will diminish the tension in the end *B* of the rope, and thus prevent a further increase in the friction. The same device can be used for a band brake of the ordinary construction. Where space below the wheel is limited, a cross bar, *C*, supported by a chain tackle exactly at its center point, may be used as shown in Fig. 726, thereby causing the action of the weight on the brake to be upward. A safety stop should be used with either form, to prevent the weights being accidentally raised more than a certain amount.

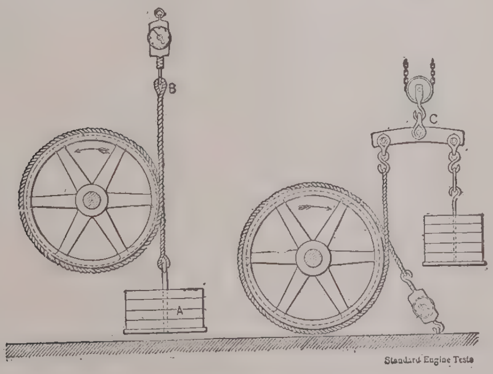

FIG. 726.

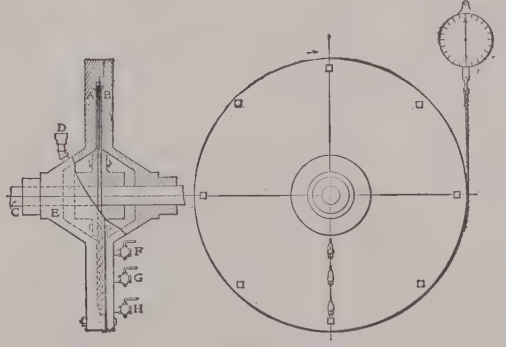

FIG. 727.

The water-friction brake is especially adapted for high speeds and has the advantage of being self-cooling. The Alden brake is also self-cooling and is capable of fine adjustment.

A water-friction brake is shown in Fig. 727. It consists of two circular disks, *A* and *B*, attached to the shaft *C*, and revolving in a case, *E*, between fixed planes. The space between the disks and the planes is supplied with running water, which enters at D and escapes at the cocks F, G, and H. The friction of the water against the surfaces constitutes a resistance which absorbs the desired power, and the heat generated within is carried away by the water itself. The water is thrown outward by centrifugal action and fills the outer portion of the case. The greater the depth of the ring of water, the greater amount of power absorbed. By suitably adjusting the amount of water entering and leaving any desired power can be obtained. Water-friction brakes have been used successfully at speeds of over 20,000 r.p.m.

13. **Speed.** There are several reliable methods of ascertaining the speed, or the number of revolutions of the engine crank-shaft per minute. The simplest is the familiar method of counting a number of turns for a period of one minute with the eye fixed on the second hand of a time piece. Another is the use of a counter held for a minute or a number of minutes against the end of the main shaft. Another is the use of a reliable tachometer held likewise against the end of the shaft. The most reliable method, and the one we recommend, is the use of a continuous recording engine register or counter, taking the total reading each time that the general test data are recorded, and computing the revolutions per minute corresponding to the difference in the readings of the instrument. When the speed is above 250 r.p.m., it is almost impossible to make a satisfactory counting of the revolutions without the use of some kind of mechanical counter.

The determination of variation of speed during a single revolution, or the effect of the fluctuation due to sudden changes of the load, is also desirable, especially in engines driving electric generators used for lighting purposes. There is at present no recognized standard method of making such determinations, and if such are desired, the method employed may be devised by the person making the test and described in detail in the report.

One method suggested for determining the instantaneous variation of speed which accompanies a change of load, is as follows: A screen containing a narrow slot is placed on the end of a bar and vibrated by means of electricity. A corresponding slot in a stationary screen is placed parallel and nearly touching the vibrating screen, and the two screens are placed a short distance from the fly-wheel of the engine in such a position that the observer can look through the two slots in the direction of the spokes of the wheel. The vibrations are adjusted so as to conform to the frequency with which the spokes of the wheel pass the slots.

When this is done the observer viewing the wheel through the slots sees what appears to be a stationary fly-wheel. When a change in the velocity of the fly-wheel occurs, the wheel appears to revolve either backward or forward according to the direction of the change. By careful observations of the amount of this motion, the change of angular velocity during any given time is revealed.

Experiments that have been made with a device of this kind show that the instantaneous gain of velocity, upon suddenly removing all the load from an engine, amounted to from one-sixth to one-quarter of a revolution of the wheel.

In an engine which is governed by varying the number of explosions or working cycles, a record should be kept of the number of explosions per minute; or if the engine is running at nearly the maximum load, by counting the number of times the governor causes a miss in the explosions.

One way of mechanically recording the explosions is to attach to the exhaust pipe a cylinder and piston arranged so that the pressure caused by the exhaust gases operates against a light spring and moves a register, which is provided for automatically counting the number.

Note.—An instrument for this purpose has been devised by R. Mathot. The following description is from his book on "Modern Gas Engines and Producer Gas Plants":

The instrument, Fig. 728, is somewhat similar in form to the ordinary indicator. Its record, however, is made on a paper tape which is continuously unwound. The cylinder c is provided with a piston p, about the stem of which a spring s is coiled. A clock train contained in the chamber b unwinds the strip of paper from the roll p' and draws it over the drum p'', where the pencil t leaves the mark. The tape is then rewound on the spindle p'''. A small stylus or pencil f traces the atmospheric line on the paper as it passes over the drum p''. In order to obviate the binding of the piston p when subjected to high temperature of the explosions, the cylinder c is provided with a casing e in which water is circulated by means of a small rubber tube which fits over the nipple e'. This recorder analyzes with absolute precision the work of all engines, whatever may be their speed. It gives a continuous graphic record from which the number of explosions, together with the initial pressure of each, can be determined, and the order of their succession. Consequently the regularity or irregularity of the variations can be observed and traced to the secondary influences producing them, such as the action of the inlet and outlet valves and the sensitiveness of the governor. It renders it possible to estimate the resistance to suction and the back pressure due to expelling the burnt gases, the chief causes of loss in efficiency in high-speed engines. Furthermore, the influence of compression is markedly shown from the diagram obtained.

The recorder is mounted on the engine; its piston is driven back by each of the explosions to a height corresponding with their force; and the stylus or pencil controlled by the lever *t* records them side by side on the moving strip of paper. The speed with which this strip is unwound conforms with the number of revolutions of the engine to be tested, so that the records of the explosions are placed side by side clearly and legibly.

Their succession indicates not only the number of explosions and of revolutions which occur in a given time, but also their regularity, the number of mis-fires. The pressure of the explosions is measured by a scale connected with the recorder-spring. By employing a very weak spring which flexes at the bottom simply by the effect of the compression in the engine cylinder, it is possible to ascertain the amount of the resistance to suction and to the exhaust. It is simply sufficient to compare the explosion record with the atmospheric line, traced by the stylus *f*. By means of this apparatus, and of the records which it furnishes, it is possible analytically to regulate the work of an engine, to ascertain the proportion of air, gas, or hydrocarbon which produces the most powerful explosion, to regulate the compression, the speed, the time of ignition, the temperature, and the like.

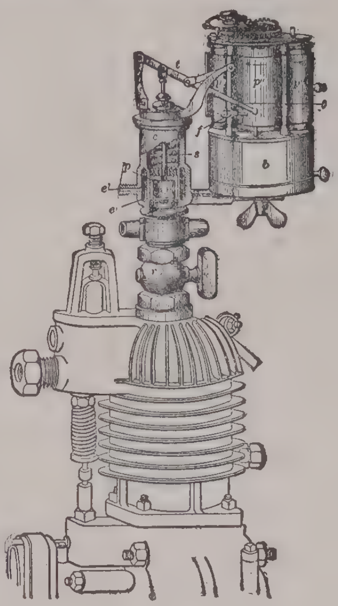

FIG. 728.

14. **Recording the Data.** The time of taking weights and every observation should be recorded, and note made of every event, however unimportant it may seem to be. The pressures, temperatures, meter readings, speeds, and other measurements should be observed every 20 or 30 minutes when the conditions are practically uniform, and at more frequent intervals if they are variable. Observations of the gas or oil measurements should be taken with special care at the expiration of each hour, so as to divide the test into hourly periods, and reveal the uniformity, or otherwise, of the conditions and results as the test goes forward.

All data and observations should be kept on suitable prepared blank sheets or in note books.

15. **Uniformity of Conditions.** When the object of the test is to determine the maximum economy, all the conditions relating to the operation of the engine should be maintained as constant as possible during the trial.

16. **Indicator Diagrams and their Analysis.** *Sample Diagrams*: Sample diagrams nearest to the mean should be selected from those taken during the trial and appended to the tables of the results. If there are separate compressions or feed cylinders, the indicator diagrams from these should be taken and the power deducted from that of the main cylinder.

17. **Standards of Economy and Efficiency.** The hourly consumption of heat, determined as pointed out in Article 9, divided by the indicated or the brake horse-power, is the standard expression of engine economy recommended.

In making comparisons between the standard for internal-combustion engines and that for steam, it must be borne in mind, that the former relates to energy concerned in the generation of the force employed, whereas in the steam engine it does not relate to the entire energy expended during the process of combustion in the steam boiler. The steam engine standard does not cover the losses due to combustion, while the internal-combustion engine standard, in cases where a crude fuel such as oil is burned in the cylinder, does cover these losses. To make a direct comparison between the two classes of engines considered as complete plants for the production of power, the losses in generating the working agent must be taken into account in both cases and the comparison must be on the basis of the fuel used; and not only this, but on the basis of the same or equivalent fuel used in each case. In such a comparison, where producer gas is used, and the producer is included in the plant, the fuel consumption, which will be the weight of coal in both cases, may be directly compared.

The thermal efficiency ratio per indicated horse-power or per brake horse-power for internal-combustion engines is obtained in the same manner as for steam engines, and is expresesd by the fraction

$$\frac{2545}{\text{B.T.U. per H.P. per hour}}.$$

18. **Heat Balance.** For purposes of scientific research, a heat balance should be drawn which shows the manner in which the total heat of combustion is expended in the various processes concerned in the working of the engine. It may be divided into three parts: first, the heat which is converted into the indicated or brake work; second, the heat rejected in the cooling water of the jackets; and third, the heat rejected in the exhaust gases, together with that lost through incomplete combustion and radiation.

To determine the first item, the number of foot-pounds of work performed by, say, one pound or one cubic foot of the fuel is determined; and this quantity divided by 778, which is the mechanical equivalent of one British thermal unit, gives the number of heat units desired. The second item is determined by measuring the amount of cooling water passed through the jackets, equivalent to 1 lb. or 1 cu.ft. of fuel consumed, and calculating the amount of heat rejected, by multiplying this quantity by the difference in the sensible heat of the water leaving the jacket and that entering. The third item is obtained by the method of differences; that is, by subtracting the sum of the first two items from the total heat supplied. The third item can be subdivided by computing the heat rejected in the exhaust gases as a separate quantity. The data for this computation are found by analyzing the fuel and the exhaust gases, or by measuring the quantity of air admitted to the cylinder in addition to that of the gas or oil.

19. **Report of Test.** The data and results of a test should be reported in the manner outlined in one of the following tables, the first of which gives a complete summary when all the data are determined, and the second is a shorter form of report in which some of the minor items are omitted.

20. **Temperatures Computed at Various Points of the Indicator Diagram.** The computation of temperatures corresponding to various points in the indicator is, at best, approximate. It is possible only where the temperature of one point is known or assumed, or where the amount of air entering the cylinder along with the charges of gas or oil, and the temperature of the exhaust gases, is determined.

If the amount of air is determined for a gas engine, together with the necessary temperatures, so that the volume and the temperature of the air entering the cylinder per stroke, and that of the gas are known, we may, by combining this with the other data, compute the temperature for a point in the compression curve. In this computation we must allow for the volume of the exhaust gases remaining in the cylinder at the end of the stroke. The temperature at the point in the compression curve where it meets or crosses the atmospheric line will be given by the formula:

$$T=\frac{491.4\,V'}{V''+V'''+V''''}-459.4; \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad (A)$$

where V' is the total volume corresponding to the point where the compression curve meets or crosses the atmospheric line; V'' the volume of the air at atmospheric pressure entering the cylinder during each working cycle, reduced to the equivalent volume at 32° F.; V''' the volume of the gas consumed per cycle reduced to the equivalent at atmospheric pressure and 32° F.; and V'''' the volume of the exhaust gases retained in the cylinder reduced to the same basis. To reduce the actual volumes to those at 32° F., multiply by the ratios of $491.4 \div (T'+459.4)$, where T' is the observed temperature of the air and of the gas used as fuel. For the exhaust gases retained in the cylinder at the end of the stroke T' may be taken as the temperature of the exhaust gases leaving the engine, provided the engine is not of the "scavenging" type.

Having determined the temperature of a point in the compression curve, the temperature of any point in the diagram may be found by the equation

$$T_1=(T+459.4)\frac{P_1V_1}{PV}-459.4. \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad (B)$$

Here T_1 is the desired temperature of any point in the diagram where the absolute pressure is P_1 and the total voulme V_1, and P and V are the corresponding quantities for the point in the compression line having the temperature T computed from the formula (A).

Formula (B) holds only where the weight of the gases contained in the cylinder is constant. It is also assumed in this formula that the density of the gas compared to air at the same temperature and pressure is the same before and after the explosion.

A second method may be employed, provided the air which enters the cylinder is measured. This will allow for any difference in the density of the gas before and after explosion, and more exact values for temperatures on the expansion curve may be obtained than by the first method.

In this method the density of the exhaust gases compared to air at the same temperature and pressure is computed, assuming perfect combustion, and including the effect of the water vapor present; and from this density the volume of the gases exhausted per cycle is determined. If this volume exhausted per cycle, added to the volume of the gas retained in the clearance space at the end of the stroke, be called V in equation (B), and T be the observed temperature of the exhaust gases, this equation may be used for determining the temperature of any point in the diagram in the way already described. This method is more complicated than the first, as it involves the determination of the theoretical density after explosion, but it possesses the advantage that it may be applied to an oil as well as to a gas engine.

A third method of computing the temperature of the various points in the diagram may be employed where analyses of the exhaust gases as well as of the fuel have to be made. This method is more complicated than the first, but, in common with the second, it possesses the advantage that it may be applied to an oil as well as to a gas engine.

In applying the third method the volume of the exhaust gases discharged per working cycle would be given by the formula:

$$V_2=\frac{1}{D}(Rw+w), \quad . \ . \ . \ . \ . \ . \ . \ . \ . \ . \ . \ . \ . \quad (C)$$

where D is the density of the exhaust gases at their observed temperature, computed from the analysis, assuming the vapor of water produced through burning the hydrogen in the fuel to be in a gaseous state; R the weight of the air which enters the cylinder per pound of fuel consumed per working cycle; the value of R, providing there are no unconsumed hydrocarbons, may be computed by employing the formula:

$$R=\frac{\mathrm{NC}}{.33(\mathrm{CO}_2+\mathrm{CO})}, \quad . \ . \ . \ . \ . \ . \ . \ . \ . \ . \ . \ . \ . \quad (D)$$

where N, CO_2 and CO represent the proportions, by volume, of the several constituents of the exhaust gases, and C the weight of carbon consumed and converted to CO_2 or CO per pound of fuel burned, computed from the analysis of the fuel and of the exhaust gases.

Having determined the volume V_2 of the exhaust gases, formula (B) may be used in computing the temperature, in which case T will represent the temperature of the exhaust gases, as in the second method, P the pressure of the exhaust, and V the volume of the exhaust gases V_2 discharged per stroke, added to the volume of the gases retained in the cylinder at the end of the stroke.

The value of R given in equation (D) is approximate, on account of the fact that the percentage of N should be that due to the air alone, and not that due to the air in addition to that contained in the fuel gas. Where extreme accuracy is desired, the value found for R may be used to determine the percentage of N which in the analysis of the exhaust gases is due to the N in the fuel gas, and this value may be subtracted from the total N shown by the analysis of the fuel gases, in order to obtain the correct value of N to be used in equation (D).

TABLE 150

DATA AND RESULTS OF TEST OF GAS OR OIL ENGINE

Arranged according to the Complete Form advised by the Engine Test Committee, American Society of Mechanical Engineers. Code of 1902.

1. Made by .... of ....
   on engine located at..
   to determine...
   ....
2. Date of trial...
3. Type of engine, whether oil or gas..
   ....
4. Class of engine (mill, marine, motor for vehicle, pumping, or other).................................
   ....
5. Number of revolutions for one cycle, and class of cycle....................................
   ....
6. Method of ignition..
   ....
7. Name of builders...
8. Gas or oil used..
   (*a*) Specific gravity.. deg. Fahr.
   (*b*) Burning-point.. "
   (*c*) Flashing-point... "
9. Dimensions of engine:

1st Cyl. 2d Cyl.

   (*a*) Class of cylinder (working or for compressing the charge)..................
   (*b*) Vertical or horizontal..
   (*c*) Single- or double-acting...
   (*d*) Cylinder dimensions...
      Bore, in...
      Stroke, ft...
      Diameter piston rod, in. ....
      Diameter tail rod, in..
   (*e*) Compression space or clearance in per cent of volume displaced by piston per stroke..
      Head end...
      Crank end..
      Average..
   (*f*) Surface in square feet (average)..
      Barrel of cylinders..
      Cylinder heads..
      Clearance and ports..
      Ends of piston...
      Piston rod...
   (*g*) Jacket surfaces or internal surfaces of cylinder heated by jackets, in square feet
      Barrel of cylinder...
      Cylinder heads..
      Clearance and ports..
   (*h*) Horse-power constant for 1 lb. M.E.P., and one revolution per minute........
10. Give description of main features of engine and plant, and illustrate with drawings of same given on an appended sheet. Describe method of governing. State whether conditions were constant throughout the test.

Total Quantities

11. Duration of test .... hours.
12. Gas or oil consumed.. cu.ft. or lbs.
13. Air supplied in cubic feet... cubic feet.
14. Cooling water supplied to jackets... "
15. Calorific value of gas or oil by calorimeter test, determined by.................. B.T.U.

Hourly Quantities

16. Gas or oil consumed per hour.. cu.ft. or lbs.
17. Cooling water supplied per hour.. lbs.

Table 150—DATA AND RESULTS OF TEST OF GAS OR OIL ENGINE—*Continued*

Pressures and Temperatures

18. Pressure at meter (for gas engine) in inches of water	ins.
19. Barometric pressure of atmosphere:	
(*a*) Reading of height of barometer	"
(*b*) Reading of temperature of barometer	deg. Fahr.
(*c*) Reading of barometer corrected to 32° Fahr	ins.
20. Temperature of cooling water:	
(*a*) Inlet	deg. Fahr.
(*b*) Outlet	"
21. Temperature of gas at meter (for gas engine)	"
22. Temperature of atmosphere:	
(*a*) Dry-bulb thermometer	"
(*b*) Wet-bulb thermometer	"
(*c*) Degree of humidity	per cent
23. Temperature of exhaust gases	deg. Fahr.
How determined	

Data Relating to Heat Measurement

24. Heat units consumed per hour (lbs. of oil or cu.ft. of gas per hour multiplied by the total heat of combustion)	B.T.U.
25. Heat rejected in cooling water:	
(*a*) Total per hour	"
(*b*) In per cent of heat of combustion of the gas or oil consumed	per cent
26. Sensible heat rejected in exhaust gases above temperature of inlet air:	
(*a*) Total per hour	B T.U.
(*b*) In per cent of heat of combustion of the gas or oil consumed	per cent
27. Heat lost through incomplete combustion and radiation per hour:	
(*a*) Total per hour	B.T.U.
(*b*) In per cent of heat of combustion of the gas or oil consumed	per cent

Speed, Etc.

28. Revolutions per minute	rev.
29. Average number of explosions per minute	
How determined	
30. Variation of speed between no load and full load	rev.
31. Fluctuation of speed on changing from no load to full load measured by the increase in the revolutions due to the change.	

Indicator Diagrams

	1st Cyl.	2d Cyl.
32. Pressure in lbs. per sq.in. above atmosphere:		
(*a*) Maximum pressure		
(*b*) Pressure just before ignition		
(*c*) Pressure at end of expansion		
(*d*) Exhaust pressure		
33. Temperatures in deg. F. computed from diagrams:		
(*a*) Maximum temperature (not necessarily at maximum pressure)		
(*b*) Just before ignition		
(*c*) At end of expansion		
(*d*) During exhaust		
34. Mean effective pressure in lbs. per sq.in		

Power

35. Power as rated by builders:	
(*a*) Indicated horse-power	H.P.
(*b*) Brake	"
36. Indicated horse-power actually developed:	
First cylinder	"
Second cylinder	"
Total	"
37. Brake H.P., electric H.P., or pump H.P., according to the class of engine	"
38. Friction indicated H.P. from diagram, with no load on engine and computed for average speed	"
39. Percentage of indicated H.P. lost in friction	per cent

TABLE 150—DATA AND RESULTS OF TEST OF GAS OR OIL ENGINE—*Continued*

Standard Efficiency Results

40. Heat units consumed by the engine per hour:
    (*a*) Per indicated horse-power .... B.T.U.
    (*b*) Per brake horse-power .... "
41. Heat units consumed by the engine per minute:
    (*a*) Per indicated horse-power .... "
    (*b*) Per brake horse-power .... "
42. Thermal efficiency ratio:
    (*a*) Per indicated horse-power .... per cent
    (*b*) Per brake horse-power .... "

Miscellaneous Efficiency Results

43. Cubic feet of gas or lbs. of oil consumed per H.P. per hour:
    (*a*) Per indicated horse-power ....
    (*b*) Per brake horse-power ....

Heat Balance

44. Quantities given in per cents of the total heat of combustion of the fuel:
    (*a*) Heat equivalent of indicated horse-power .... per cent
    (*b*) Heat rejected in cooling water .... "
    (*c*) Heat rejected in exhaust gases and lost through radiation and incomplete combustion "
    Sum = 100 "
    Subdivisions of Item (*c*):
    (c_1) Heat rejected in exhaust gases .... "
    (c_2) Lost through incomplete combustion .... "
    (c_3) Lost through radiation, and unaccounted for .... "
    Sum = Item (*c*) ....

Additional Data

Add any additional data bearing on the particular objects of the test or relating to the special class of service for which the engine is to be used. Also give copies of indicator diagrams nearest the mean and the corresponding scales. Where analyses are made of the gas or oil used as fuel, or of the exhaust gases, the results may be given in a separate table.

TABLE 151

DATA AND RESULTS OF STANDARD HEAT TEST OF GAS OR OIL ENGINE

Arranged according to the Short Form advised by the Engine Test Committee, American Society of Mechanical Engineers. Code of 1902

1. Made by .... of ....
   on engine located at ....
   to determine ....
   ....
2. Date of trial ....
3. Type and class of engine ....
   ....
4. Kind of fuel used ....
   (*a*) Specific gravity .... deg. Fahr.
   (*b*) Burning-point .... "
   (*c*) Flashing-point .... "
5. Dimensions of engine:

	1st Cyl.	2d Cyl.
(*a*) Class of cylinder (working or for compressing the charge)		
(*b*) Single- or double-acting		
(*c*) Cylinder dimensions:		
Bore, in		
Stroke, ft		
Diameter piston rod, in		
(*d*) Average compression space, or clearance in per cent		
(*e*) Horse-power constant for 1 lb. M.E.P. and one revolution per minute		

Table 151—DATA AND RESULTS OF STANDARD HEAT TEST OF GAS OR OIL ENGINE—*Continued*

Total Quantities		
6. Duration of test	hours	
7. Gas or oil consumed	cu.ft. or lbs.	
8. Cooling water supplied by jackets.		
9. Calorific value of fuel by calorimeter test, determined by ....calorimeter	B.T.U.	
Pressures and Temperatures		
10. Pressure at meter (for gas engine) in inches of water	ins.	
11. Barometric pressure of atmosphere:		
(*a*) Reading of barometer	"	
(*b*) Reading corrected to 32° Fahr.	"	
12. Temperature of cooling water:		
(*a*) Inlet	deg. Fahr.	
(*b*) Outlet	"	
(*c*) Degree of humidity	"	
13. Temperature of gas at meter (for gas engine)		
14. Temperature of atmosphere:		
(*a*) Dry-bulb thermometer	deg. Fahr.	
(*b*) Wet-bulb thermometer	"	
15. Temperature of exhaust gases	"	
Data Relating to Heat Measurement		
16. Heat units consumed per hour (pounds of oil or cubic feet of gas per hour multiplied by the total heat of combustion)	B.T.U.	
17. Heat rejected in cooling water per hour	"	
Speed, Etc.		
18. Revolutions per minute	rev.	
19. Average number of explosions per minute		
Indicator Diagrams		
20. Pressure in lbs. per sq.in. above atmosphere:	1st Cyl.	2d Cyl
(*a*) Maximum pressure		
(*b*) Pressure just before ignition		
(*c*) Pressure at end of expansion		
(*d*) Exhaust pressure		
(*e*) Mean effective pressure		
Power		
21. Indicated horse-power:		
First cylinder	H.P.	
Second cylinder	"	
Total	"	
22. Brake horse-power	"	
23. Friction horse-power by friction diagrams	"	
24. Percentage of indicated horse-power lost in friction	per cent	
Standard Efficiency and Other Results		
25. Heat units consumed by the engine per hour:		
(*a*) Per indicated horse-power	B.T.U.	
(*b*) Per brake horse-power	"	
26. Pounds of oil or cubic feet of gas consumed per hour:		
(*a*) Per indicated horse-power	lbs. or cu.ft.	
(*b*) Per brake horse-power	"	

Additional Data

Add any additional data bearing on the particular objects of the test or relating to the special class of service for which the engine is to be used. Also give copies of indicator diagrams nearest the mean, and the corresponding scales.

2. Rules for Testing Gas Producers and Gas Engines

(Code of the German Society of Engineers[1]).

(*All metric units have been transposed to English units.*)

The preparation of the following rules for making gas engine and producer tests was undertaken by a committee appointed from the Verein Deutscher Ingenieure, in collaboration with the German Society of Engine Builders, with the view of establishing definite general regulations governing such tests. It is desirable, by specifying the important proportions of the examined plants and the conditions under which the results were obtained, to insure that these results are not only applicable to a single case, but that they have general value. To attain this end it is necessary that all data should be given uniformly according to a code of regulations such as that here presented.

The execution of such tests should be intrusted only to persons possessing the required expert knowledge and practical experience. These persons must make a trial plan, or schedule, appropriate to the individual case in hand, which, in many instances, will not require that all of the investigations stipulated in the general code are actually carried out. They must further examine the instruments for measuring or recording purposes as to their fitness and must compile the results. The following rules, the adoption or selection of which must be left to the soundness of judgment of the investigator, are intended to serve as a basis on which to proceed.

GENERAL REGULATIONS

Object of Investigation.

1. The object of a test made on a producer-gas plant may be to determine:
   - (*a*) The quantity, composition, and calorific value of the fuel consumed.
   - (*b*) The quantity, composition, and heat value of the gas produced.
   - (*c*) The degree of efficiency of the producer-gas plant.
   - (*d*) The separate heat losses in the plant.
   - (*e*) The quantity of impurities contained in 1 cm. or 1 cu.ft. of gas (dust, tar, sulphur, etc.).
   - (*f*) The moisture contents of the gas.
   - (*g*) The water consumption of the producer-gas plant, either total or in the separate parts.
   - (*h*) The mechanical work required for operating the plant, including apparatus.
   - (*i*) The duration of time required for starting.
   - (*k*) The stand-by losses during intervals of shutting down day or night.
2. The object of a test made on an internal-combustion (gas) engine may be to determine:
   - (*a*) The indicated capacity and the effective output.
   - (*b*) The mechanical efficiency.
   - (*c*) The fuel consumption and the heat consumption per horse-power hour.
   - (*d*) The consumption of lubricants, separately for cylinder and engine.
   - (*e*) The consumption of water and the heat conducted to the cooling water.
   - (*f*) The fluctuations in number of revolutions.
   - (*g*) The composition of exhaust gases.

NUMBER AND DURATION OF TESTS.

Admissible Fluctuations.

3. The number and duration of trials are determined by the purpose of the test as well as by a consideration of the conditions of installation and operation, and must be settled and previously arranged according to paragraphs 4 to 8. For trials of special importance the results of which are decisive for acceptance tests, for penalties or for premiums, this item deserves special consideration.

4. Acceptance tests should be made if possible immediately after a plant has been put into actual operation; the manufacturers, however, must be granted a reasonable time for making

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[1] Mainly from F. E. Junge's translation in *Power*, Feb., 1907.

preliminary trials of their own and for carrying out alterations or improvements when necessary. The length of this time and other conditions are best agreed upon when drawing up the delivery contract.

5. In order to be able to get acquainted with the operation of the plant that is to be tested, to find time for examining the testing devices employed, and to break in observers and assistants, it is desirable that preliminary trials be allowed.

6. If the fuel consumption in gas producers is to be determined, the trial run must be extended over at least eight hours under constant conditions and without interruptions.

7. For determining the consumption of liquid or gaseous fuel and provided the conditions are constant, it is sufficient for the higher loads to extend measurements over an hour, while for finding the consumption at the lower loads, measurements of even shorter duration are sufficient. To ascertain the constancy of the conditions the temperature of the outflowing cooling water must be read from time to time. These rules as to the duration of the tests are made with the provision that no interruption or disturbance of the trial takes place, and that intermediate readings show only slightly varying values for the consumption.

8. If only the mechanical efficiency of an engine is to be determined, trials of short duration under constant conditions are sufficient; but at least ten sets of indicator cards should be taken.

9. For investigations of special importance at least two tests should be made, one after the other. They should be accepted only if no interruptions occurred and if the results show no greater deviations than those due to unavoidable errors of observation. The mean of the two results is to be taken as the final result.

10. The extent to which the capacity and the consumption of gas may differ from the guarantee or contract figures, without justifying a claim of breach of contract, is to be clearly stated before the tests (either in the original contract or in the schedule of tests). When no other agreement has been previously arrived at, the capacity guarantee is regarded as fulfilled if the figure obtained in the test is not more than 5% below the value on which the guarantee was based. This margin, however, is allowable only for the maximum output which was promised beyond the guaranteed continuous output. The latter must be rendered by the engine under all circumstances.

The consumption of fuel and water as determined on test should not exceed the guaranteed figures by more than 5% even if, during the trial, the engine load fluctuated somewhat from the load upon which the guarantee was based, provided that fluctuation do not exceed an average of ±5% of such load, or a maximum of ±15%.

Since it is often impossible when making tests to have the internal combustion engine work at exactly the effective (horse-power) capacity on which the guarantee agreed upon in the contract is based, it is recommended that the agreement shall specify the expected fuel consumption for higher and lower outputs. The same provision is preferably made also with gas producers.

UNITS OF MEASUREMENT AND DESIGNATIONS

11. When giving pressure data it must be stated whether absolute pressures or gauge pressures above or below the atmospheric are meant. Absolute pressure equals atmospheric pressure plus gauge pressure.

12. All temperature and heat measurements refer to the Fahrenheit scale.

13. The mechanical equivalent of heat is taken at 778 ft.-lbs.

14. The calorific value of a fuel is to be taken as its lower heating value; that is, the heat which is liberated by the complete combustion of the fuel when the burnt products are cooled down to the original (room) temperature at constant pressure, it being assumed, however, that the water of combustion and the moisture contained in the fuel remain vaporized. The calorific value must be based on the unit quantity or weight of original fuel, without deducting ash, moisture, etc., and is to be expressed in heat units. For both solid and liquid fuels the unit of weight is the pound.

The heat value of gaseous fuels is based on 1 cu.ft. at 32° F., and 760 mm. barometer pressure, or must be expressed in thermal units as "effective" heat value, that is, reduced to 1 cu.ft. of actual gas used. If not specially stated, it is always understood that the heat value recorded is that of gas at 32° F. and 760 mm. barometer pressure.

In this country the general standard so far recommended seems to indicate for "standard gas" a temperature of 60° F., and a pressure of 14.7 lbs. per sq.in., corresponding to the usual atmospheric pressure.

15. The efficiency of a gas-producer plant is the ratio of the latent heat contained in the gas as produced to the heat of combustion of the total weight of fuel consumed in the plant, both items being computed from the lower heating value. In producer-gas plants having a separately fired steam boiler, it is advisable also to determine the ratio of the heat which is chemically bound in the producer gas to the heat equivalent of that portion of the fuel which is consumed in the producer proper for making such gas.

16. The unit of measurement used for the power or work output of an internal-combustion engine is the horse-power equal to 33 000 ft.-lbs. per minute. It must be clearly stated whether the indicated power, or the useful or available power, is meant. If not otherwise designated it is understood that the figures refer to the useful or available output.

17. The indicated power of the engine, or the indicated work, is the difference between the total power developed or work done, and the indicated power, or work, which is consumed within the engine; in short, the difference between the positive and the negative indicated power or work.

Note. This is the provision which caused considerable discussion among gas-engine experts some time ago. It means, as it stands, that in a 4-cycle machine, the indicated horse-power is that determined from the work diagram minus the work shown by the lower loop diagram; and, in a 2-cycle engine, the total indicated horse-power, as determined from the diagram of the power cylinder minus the pump work, is considered as the indicated horse-power. This view is undoubtedly correct when the mechanical efficiency of the engine itself as a machine is to be determined.

The power required at "no load" is the power indicated when no useful work is rendered by the engine.

18. Mechanical efficiency is the ratio of the useful power to the indicated power of the engine.

19. All consumption figures should be reduced to the hour basis, and if they are to be compared with the output of the engine they must be based on one horse-power hour. If not otherwise agreed upon, these data refer to the useful or available output at full load.

EXECUTION OF TESTS

20. If the quantity of gas made in a producer or the weight of fuel consumed in an engine is to be measured, then all pipes or ducts which are not used in the test must be cut off from the piping which leads to the producer and engine that are to be tested. This is best done by means of blind flanges. The active ducts, pipes, gas holders, etc., must be examined with regard to leakage and made tight if necessary. Unavoidable losses due to leakage must be determined. This holds especially for masonry gas mains.

FUEL CONSUMPTION OF A GAS-PRODUCER PLANT

21. The kind, number, and duration of tests must be agreed upon according to the general rules laid down in paragraphs 1 to 10.

22. The constructive features and the operative conditions of gas-producer plants must be described and illustrated in the report by drawings, so far as this is necessary, to arrive at a clear understanding of the manner of working and of the results obtained.

23. Before making the test the plant should be examined as to whether or not it is in good working order.

24. The quantity of fuel consumed in the gas producer is determined by taking the weight of the fuel which is charged into the producer during the trials in order that the producer may contain at the end of the test exactly the same amount of heat, either liberated or chemically bound in the fuel, that it contained when starting the test. To meet this requirement it is not sufficient that the depth of the fuel bed be the same at the end as it was at the beginning; it must also be taken into consideration what influence the ash and the slag left in the producer, the location of the incandescent zone, the formation of fissures and cavities, the closeness or density of the producer charge, and the chemical composition of the burning fuel particles exercise on the heat contents of the producer.

In order to comply with this requirement the following rules should be followed:

25. When starting the test the plant should be in the condition of stability or normal working condition, if possible. This means that after a period of shut-down for cleaning or

repairs it should be in active operation for one or more days, running on fuel of the same characteristics and size, with the same depth of fuel bed, the same skill of attendance as regards the charging or feeding of fresh fuel and the removing of slag, and under the same load conditions that will obtain during the test.

26. During the trial the producer should be charged and poked as nearly in accordance with the requirements for attendance as possible. The level of fuel charged must be the same at the beginning and at the end of the tests and should be kept constant during the trial. About half an hour before starting and before stopping a test, the slag and ashes should be removed.

If it is impossible to take out the ashes during the operation of the producer, the plant must be shut down immediately after stopping the test, the ashes must be taken out at once and the producer refilled up to the same level that existed when starting the test. The weight of fuel used for this purpose must be added to the consumption.

27. The fuel consumed during the trial must be weighed, also the fuel which has not been burned and remains useful; that is, that portion which drops down from above the grate while raking out the ashes, and that which is culled out form the ashes as unburned. The weight of the former may be deducted from the consumption, but not the amount which is taken out from the ashes, nor the coal dust which accumulates in the scrubbers and in the flues between the producer and the engine.

28. To be able to determine the quantity of ash and slag produced during the trial, the ash box must be emptied before the test. If this is not possible, as when an inclined grate is used, the refuse in the ash box must be equalized before and after the run.

29. The stand-by losses during intervals of shutting down at day and night must be determined.

30. In order to get a representative sample of the solid fuel, the following course may be pursued: Of every carload, basket, or other measure of fuel, put a shovelful into a covered receptacle. Immediately after the test is over, the contents of the receptacle should be broken, mixed, spread, and quartered by drawing the two diagonals of a square. The two opposite quarters are rejected, the two others broken up finer, mixed, and quartered, and the two opposite quarters rejected. This is continued until a sample of some 10 to 20 lbs. remains, which is preserved in well closed receptacles for analysis. In addition to this a number of other samples must be put away in air tight receptacles for use in determining the contents of moisture in the fuel.

31. The composition of the fuel shall be determined by chemical analysis. Its contents in carbon (C), hydrogen (H), oxygen (O), sulphur (S), ash (A), and water (W) must be given in percentage of weight referred to the original fuel. The contents, in the fuel, of nitrogen (N) can be disregarded. The behavior of the fuel when being heated should be determined by a coking test.

32. The calorific value of the fuel must be determined by calorimetric analysis. An approximate determination of the heating value can be made on the basis of the chemical analysis by employing Du Long's formula:

$$\text{Heating value} = 145\,\mathrm{C} + 522.3\left(\mathrm{H} - \frac{\mathrm{O}}{8}\right) + 40\,\mathrm{S} - 9.66\,\mathrm{W},$$

in which C, O, H, S, and W are expressed in weight per cent.

TESTING AN INTERNAL-COMBUSTION ENGINE

33. Kind, number, and duration of trials to be agreed upon according to the general regulations Nos. 1 to 8.

34. The constructive features and operative conditions of the engine must be so illustrated in the report as to enable one to form a correct idea of the manner of working and of the results of operation. Especially important are the type and capacity of engine, diameter of cylinder and piston-rod, piston stroke, contents of clearance space, and other essential dimensions; the normal rate of revolution and the admissible fluctuations; kind and heat value of fuel for which the engine is intended. The diameter of the cylinder and the stroke should be actually measured if this is possible.

The contents of the compression space are preferably determined by filling with water. If it is impossible to state the cubical contents of the compression space, then the compression pressure at full load should at least be given. This is done by taking an indicator card while the ignition is interrupted.

35. Before making the test the engine must be examined internally and externally as to whether or not it is in good working order.

36. The number of revolutions of the engines should be determined by a continuous speed counter, the records of which must be noted at certain intervals, and must be checked or corrected from time to time by direct readings. If the speed conditions of the engine are to be investigated it is essential to determine the following items:

(*a*) The number of revolutions under constant conditions at maximum load and at no load.

(*b*) The fluctuations in speed at constant load.

(*c*) The temporary change in the number of turns when the load is suddenly decreased or increased from a given constant load by a prescribed amount. These determinations can be executed with apparatus of the character of the Horn tachograph. The fluctuations of speed during the performance of one engine cycle above and below the mean value, expressed in parts of the latter, should be determined by calculation unless otherwise provided.

The coefficient of fly-wheel regulation is

$$\delta=\frac{N_{\max}-N_{\min}}{\frac{N_{\max}+N_{\min}}{2}}=2\left(\frac{N_{\max}-N_{\min}}{N_{\max}+N_{\min}}\right),$$

where N = number of revolutions.

37. The useful output can be determined either by brake test or by electrical measurement.

The dimensions and weight of the brake should be determined before the trial.

The electrical measurements can be made on a generator directly coupled to the gas engine. The useful work is computed from the output of the dynamo. The efficiency of the generator should be determined by one of the methods as laid down in the "Rules for Judging and Testing Electrical Machinery and Transformers," published by the Association of German Electrical Engineers. If the efficiency is found approximately by measuring the determinable losses, then an adequate amount (say 2% of the full load output) must be allowed for losses not accounted for.

The apparatus with which the electrical measurements are executed must be calibrated before and if possible also after the test.

Whether anything besides the 2% above allowed should be credited to the gas engine for increased bearing friction and windage of the generator, must be settled in each individual case.

Whether, in case the useful output can neither be determined by brake test or by electrical measurements, the code provision for testing steam engines can be admitted as correct for gas engines, namely, to designate the useful output as the difference between the indicated work at any load and the indicated work at no load, cannot be settled at the present state of development, since results of accurate investigations are not yet available.

38. Indicators must be connected immediately to the combustion chamber without employing long piping with sharp bends, and one indicator must be provided for every combustion chamber. For this purpose each compression chamber must have an opening of $\frac{3}{4}$ or 1 in. Whitworth thread. The same holds true for pump cylinders.

The indicators and their springs must be calibrated before and after the test according to the accepted standards.

39. During the test, cards should be taken quite frequently from every combustion chamber and from the pump cylinders. The cards should be designated by numbers, and the time when each card was taken, the scale of springs used and the number of single cards obtained must be recorded on the cards. At least five diagrams should be taken on one card successively. From time to time diagrams indicated with a weak spring should be taken from the combustion chambers.

The indicated work at no load should be determined immediately after stopping the main test and while the engine is still warmed up ready for operation. Care must be taken that the no load cards are not taken during an acceleration or during a retardation period of the fly-wheel.

ANALYSIS OF THE GAS GENERATED IN A PRODUCER GAS PLANT OR CONSUMED IN AN INTERNAL-COMBUSTION ENGINE, OR OF THE LIQUID FUEL USED

40. The samples for the chemical analysis of the gas must be taken during the trial at regular intervals and as frequently as possible.

They must be either analyzed on the spot or preserved in glass tubes closed by melting the ends. The analysis is to determine, in per cent of volume, the contents of the gas in carbon monoxide (CO), carbon dioxide (CO_2), hydrogen (H_2), marsh gas (CH_4), heavy hydrocarbons and oxygen (O_2).

In addition it is recommended to determine the contents of sulphur. The gas samples should be taken from the gas main between the cleaning apparatus and the engine.

41. The heat value of the gas should be determined as often as possible by calorimetric analysis, and the burner of the calorimeter should be fed from the gas main without interruption. In suction producer plants this can be done by means of a gas pump drawing from the main. If conditions should make it necessary that a sample be taken from the pipe while the calorimeter is shut off, such sample to be later transferred to and burned in the calorimeter, then the quantity of gas so taken should not be less than 300 liters (10 cu.ft.), in order that the calorimeter may at first be brought into the condition of stability as regards the water of combustion, and in order that at least 100 liters (3.5 cu.ft.) remain available for two successive analyses. The suction pump, the gas holder and the piping must be made tight with special care when making a calorimeter analysis of suction gas.

42. The gas meter of the calorimeter in which the heat value of the gas is determined must be calibrated. For determining the temperatures of the calorimeter water, only thermometers with calibration certificates or others compared with such should be used. The scales must be divided at least into tenths of a degree.

On the basis of the chemical analysis the heating value per standard cubic foot of gases which do not contain heavy hydrocarbons can be computed from the following formula, if a calorimetric analysis cannot be made

$$\text{Heating value} = 3.42\ CO + 2.97\ H_2 + 9.52\ CH_4$$

where CO, H_2 and CH_4 are expressed in volume per cent.

43. The quantity of gas produced or consumed should be measured by means of a gas holder or a gas meter. The cross-sectional area of the holder should be determined by measurement of its circumference at several places. Consumption tests with the gas holder shall not be made while the latter is exposed to the sun.

44. The gas meter must be calibrated and set level; it must be so filled that the water level corresponds to the normal filling existing during calibration. Between the gas meter and the engine a pressure regulator must be installed or a large suction space provided so that the water level shows only small pulsations during the pressure fluctuations.

45. At intervals corresponding to the duration of test the following readings should be taken: Position of the bell of the gas holder at three places or the records shown by the gas meter; the pressure in the bell or in the gas meter; the temperature of the gas when entering and when leaving the gas holder or the gas meter and before reaching the engine; the barometric pressure.

46. If the temperature of the gas is different when measuring the consumption than when measuring the heat value, the computation must also take into account the increase of volume which is due to the moisture contents of the gas at higher temperatures.

47. The consumption of liquid fuel must be determined either by weight or by measuring its volume. For determining heat value, composition, and specific weight of the fuel one representative average sample is sufficient.

48. When measuring the fuel consumption of internal-combustion engines, the consumption of lubricating oil for the cylinder should be determined at the same time.

49. If the consumption at low loads of a double-acting tandem or twin engine is to be determined, it is not allowable to shut off the gas from one or more ends of the cylinders, provided that no other arrangements have been previously agreed upon and are mentioned in the report, or that the governor acts automatically in the way described.

EXPLANATIONS TO VARIOUS ARTICLES OF THE CODE

The main code is followed by a number of explanations from which the following extracts are taken. The figures refer to the paragraphs of the above code.

1 and 2. In most cases only one or two of the objects of test mentioned are taken into account in any given trial. If in any exceptional case the object of the test should not be any of those mentioned, it should be a simple matter to adapt the rules given.

Under 2 (*c*) the term horse-power hour is used. It is essential that in any given trial this term be more closely defined, as horse-power may mean indicated brake, or even horse-power developed by pumps.

4. It is extremely desirable that the contract state the time allowed the manufacturer for adjustment and trial runs, because his own interests may make him call sometimes for a long, sometimes for a short period. In the case of a small engine, more or less a commercial stock machine, he may wish to have the period as short as possible, and this the buyer may agree to without danger of loss to himself. If, however, the machine is of a special type, or one provided with special attachments, it is but a matter of justice to allow the manufacturer a reasonable time in which to break in the engine and to give him an opportunity to correct any imperfections that may appear. It is to the interest of the buyer to grant such a period in order to become familiar with the machine before taking over the entire responsibility of operating it. It is also true that many faults appear only after some weeks of operation.

On the other hand, too long a period of adjustment is in many cases not acceptable to the buyer, because any extended work of improvement usually seriously hampers operation; and because in many cases he desires an operative machine, which no longer requires the care of the manufacturer, as soon as possible.

It frequently happens that no acceptance test is agreed upon. In such cases it sometimes happens that the buyer comes back upon the manufacturer for faults which did not develop until the machine had been in operation some time.

If the manufacturer then agrees to an investigation or a test, a sufficient period should be given him to make any investigation he sees fit or to correct any imperfections that may have appeared before the decisive trial or investigation is made. This sometimes leads to a simple settlement of the matter in that the manufacturer discovers that ignorance or carelessness on the part of the operator have caused the imperfections complained of. The granting of such a period also guards the buyer against any later claim of the manufacturer that during the trial the machine was not in the condition in which he delivered it.

5. Preliminary tests are always desirable, but not absolutely necessary. The cost of any kind of investigation is usually quite high and of course the cost increases directly with the time. The expert called will therefore make such tests when they seem to him essential. But the manufacturer should have the right to call for the time necessary for such trials if he is to present the machine in its best condition.

6. It cannot be denied that eight hours is a rather short time, because it is extremely difficult to determine whether the producer is in the same condition at the end as at the beginning of the test, and because this uncertainty may lead to large errors. On the other hand, it is unquestionable that in many cases a longer time would call forth so many difficulties in operation that eight hours would seem the necessary limit.

The rule is mainly framed to prevent trials of so short duration that serious errors can hardly be avoided, but it leaves it to the judgment of the experimenter whether to make the tests longer than eight hours where it seems desirable and is possible to do so.

7. Intermediate readings are recommended without qualification, since they form the best criterion of the constancy of conditions. With liquid or gas fuels of constant composition, individual readings every five minutes apart sometimes show no variation for hours at a time. In such a case it is useless to extend the time of the trial.

8. In determining the mechanical efficiency of an engine it should not be forgotten that, although the average load may be constant, there may be speed variations due to the inevitable inequality of the power impulses, so that during some cycles work is done in accelerating the fly-wheel, while during others the fly-wheel by retardation gives up some of its kinetic energy.

To minimize any error that this may introduce into the determination of the mechanical efficiency, at least ten indicator diagrams should be taken.

If the conditions are otherwise constant, however, it is not necessary to spread these diagrams over any considerable period of time.

It is self-evident that during the time of taking the diagrams the supply of lubricating oil must not be increased.

Changes in the mechanical efficiency of the engine, as for instance those due to fouling, cannot be detected with certainty even by a long test period; they become noticeable usually only after a period of operation extending over two weeks. The determination of the mechanical efficiency of an engine, after constant conditions of operation are attained, therefore only applies to the engine in its then existing state or condition.

The number of diagrams to be taken on one card cannot be definitely stated. On account of variation in the diagrams, which is less at high than at low loads, care should be had not to take too few. On the other hand, it is useless to take more than can be clearly distinguished. The running together of a larger number of diagrams only makes their evaluation more uncertain.

10. In consideration of unavoidable errors of observation, possible errors of the instruments used, etc., it is meet and usual to allow a certain margin between the figures found on trial and those guaranteed. In the steam engine code 5% is allowed for this, and it seems reasonable to assume the same figure in this case. Only in one point, in the guaranteed normal capacity, does the gas engine call for an exception.

A given steam engine gives its most economical results at a certain cut-off, but a higher capacity can always be obtained at the expense of a little economy, that is, a buyer is certain that even a machine slightly too small will give him sufficient capacity. A gas engine, on the contrary, works with the greatest economy at its maximum load. It is to the interest of the buyer, therefore, to get an engine exactly suited to his needs and not to choose it too large. It is possible for the same reason that any engine, if lacking slightly in guaranteed capacity, may become absolutely useless to the buyer. For these reasons it was thought advisable not to grant the manufacturer any leeway whatever as regards guaranteed capacity.

It is clear, therefore, that the manufacturer must take upon himself any possible inaccuracies in the measurements, unless he can show them up and demand a new trial. For that reason it is well for him to make his guarantee a little on the safe side of what he knows his engine is capable of developing. On the other hand, there is no harm done to the interest of the buyer if the manufacturer underrates the normal capacity of his machine, because the former will always call for an engine of a certain normal capacity to suit his needs. If he fails to do this, but places his dependence in the guaranteed maximum capacity, he is open to the charge of carelessness.

Since during acceptance tests it is often not possible to keep the load quite constant, it became necessary, following the steam engine code, to allow a certain amount of variation, within which no just cause could be found for objection to the trial. There are cases where the variations occurring are much greater, as when a gas engine is used for driving a roll train. But no one set of rules can possibly take into account all such extreme cases, and in such instances the contract should contain the necessary agreements to make any test clear and free from subsequent objections.

The wish has been expressed from several quarters, that the rules should contain a definition of the term "normal capacity." On account of the peculiarity of the gas engine above discussed, this is not quite feasible. But the term "maximum continuous capacity" perhaps defines most nearly what is intended in most cases.

14. It is sometimes the case that the heating value of the standard cubic foot, that is, reduced to 32° F. and 760 mm. barometer, is so greatly different from the actual value of the gas as used, that any contract which contains only the heating value of the gas stated on that basis does not convey much meaning to the non-technical buyer. If for instance a given gas has a heating value of 135 B.T.U. per standard cu.ft., its effective heating value at a high altitude and in a warm climate, say as 68° and 620 mm. barometer, will only be about 100 B.T.U. per cu.ft. To obviate any misunderstanding, it should be clearly stated that, when the effective heating value of the gas is not definitely specified, the heating value at 32° F. and 760 mm. barometer is meant.

19. By full load is meant the normal capacity, as per paragraph 10.

23. For acceptance tests, and all other tests which are intended to decide any disagreements between manufacturer and buyer, such examination should be carried out in the presence and with the aid of the former, as already mentioned under paragraph 4.

24–26. In all gas producer tests it is hardly possible with certainty to have all conditions exactly the same at the end as at the beginning. But since any difference in the beginning and end conditions may lead to considerable error, which can only be equalized by excessive length of test, the rules are intended to operate to the end that such errors are not in any way magnified by the method of test. Hence the detailed statement in the regulations.

27. Since in actual operation the fuel in the ash or the coal dust in the gas mains are hardly ever utilized, no correction should be made for these on any trial. In order to prevent, however, the results from being influenced by insufficient cleaning of the producer, any fuel which falls out from above the grate during the cleaning period may be subtracted from the amount charged.

35. See explanation under 23.

37. A brake test of a large engine is in some instances not possible, and in any case a matter of considerable cost. In many cases, however, the larger gas engines are either direct connected to a generator or to some other power consumer, as a blowing cylinder. In the former case electrical measurements, from which the effective horse-power may be determined, are easily made. In the latter case the capacity guarantee will in most instances be based upon the performance of the power consumer, as for example the air compressed by the blowing cylinder. Outside of engines of this type, however, there still remain many cases in which it would be of the utmost value to have some means of determining the effective capacity, and it should not be forgotten that, even in the case of medium-sized machines, a braking of the engine at the place of erection is often, on account of local restrictions, very difficult. The problem has been solved for steam engines by assuming that the difference between the indicated horse-power at any load and the indicated horse-power at no load is the effective or useful horse-power. It is quite possible that in many cases this is not quite correct, but the method is very generally accepted and followed.

On account of the great overload capacity of the steam engine, a small error in this respect does not mean a great deal. But the case of the gas engine is quite different. The data on hand does not warrant the application of the same method to the gas engine, and the consequences of an erroneous conclusion are much more serious on account of the lack of overload capacity.

For these reasons one is compelled in some cases to omit the determination of the effective capacity altogether and to be content with the determination of the indicated power only. It is recommended in such cases that the mechanical efficiency be not assumed too high and that any guarantees regarding fuel, etc., also be based upon the indicated horse-power.

It is sometimes possible to brake an engine on the test floor of the factory. The mechanical efficiency may thus be previously determined when it is known that no brake test can be made in the final place of erection.

39. The number of diagrams to be taken during any given test cannot be definitely specified. Much depends upon the length of test, and the decision may be left to the judgment of the experimenter.

It is, however, always recommended that a bundle of diagrams, instead of only one, be taken on every card. Thus a series of diagrams are obtained, while, if only a single diagram is taken, it is possible to hit upon the same diagram in the series a number of times. (See under extract 8.)

The work of fluid friction, that is, the lower-loop diagram, cannot be determined with certainty from the full indicator cards. It is best for that reason to ignore the loop when determining the positive work and to find the negative work from special weak spring diagrams.

48. The measurements of the quantity of lubricating oil used is of importance in smaller engines, because the fuel consumption can be favorably influenced by a copious supply of the lubricant.

49. If under low loads, only one end of the cylinder is allowed to work, the fuel consumption would be much lower. But since this is not generally done in operation, the results would be erroneous. If, however, the governor during operation shuts off the individual cylinders or cylinder ends, as the load drops, this is of course also permissible during a test.

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